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# **INDEPENDENT ASSESSMENT REPORT (IAR): EVALUATION OF A NEW SOLAR AIR CONDITIONER**

*Prepared For:*  
**California Energy Commission**  
Public Interest Energy Research Program

*Prepared By:*  
**WorkSmart Energy Enterprises, Inc.**

**PIER FINAL PROJECT REPORT**

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## PREFACE

The Public Interest Energy Research (PIER) Program supports public interest energy research and development that will help improve the quality of life in California by bringing environmentally safe, affordable and reliable energy services and products to the marketplace.

The PIER Program, managed by the California Energy Commission (Commission), annually awards up to \$62 million of which \$2.4 million/year is allocated to the Energy Innovation Small Grant (EISG) Program for grants. The EISG Program is administered by the San Diego State University Foundation under contract to the California State University, which is under contract to the Commission.

The EISG Program conducts four solicitations a year and awards grants up to \$75,000 for promising proof-of-concept energy research.

PIER funding efforts are focused on the following six RD&D program areas:

- Residential and Commercial Building End-Use Energy Efficiency
- Industrial/ Agricultural/Water End-Use Energy Efficiency
- Renewable Energy Technologies
- Environmentally-Preferred Advanced Generation
- Energy-Related Environmental Research
- Energy Systems Integration

The EISG Program Administrator is required by contract to generate and deliver to the Commission an Independent Assessment Report (IAR) on all completed grant projects. The purpose of the IAR is to provide a concise summary and independent assessment of the grant project in order to provide the Commission and the general public with information that would assist in making follow-on funding decisions. The IAR is organized into the following sections:

- Introduction
- Objectives
- Outcomes (relative to objectives)
- Conclusions
- Recommendations
- Benefits to California
- Overall Technology Assessment
- Appendices
  - Appendix A: Final Report (under separate cover)
  - Appendix B: Awardee Rebuttal to Independent Assessment (Awardee option)

For more information on the EISG Program or to download a copy of the IAR, please visit the EISG program page on the Commission's Web site at:

<http://www.energy.ca.gov/research/innovations> or contact the EISG Program Administrator at (619) 594-1049, or email at: [eisgp@energy.state.ca.us](mailto:eisgp@energy.state.ca.us). For more information on the overall PIER Program, please visit the Commission's Web site at <http://www.energy.ca.gov/research/index.html>.

# Evaluation of a New Solar Air Conditioner

## EISG Grant # 00-16

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Grant Funding: \$74,547  
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### 1.0 Introduction

Air conditioning powered by solar energy has great potential, in part because high demand for cooling usually coincides with plentiful sunlight. The intensity of solar energy on the roof of a typical single-story building in California is roughly ten times the cooling requirement for the same building. Solar air conditioning at an economically competitive level could reduce electricity costs for residential and small commercial customers. This would cut the growth of peak electric demand and ease the increasing pressures on generating capacity, transmission, and distribution. Currently available technologies are neither practical nor cost-effective. Photovoltaic (PV) systems require a large roof area and cost many times more than a conventional air conditioner. Thermally driven absorption cooling requires costly, high-temperature collectors and undesirable cooling towers. Furthermore, these systems have a disconnect of several hours between peak cooling capacity and peak cooling demand. That in turn requires electric or thermal storage in order to maximize the solar contribution.

A solar air-conditioning system employing relatively inexpensive low-temperature collectors, coupled with an innovative desiccant dehumidification and evaporative process, provides a new prospect for cost-effective solar cooling. If proven practical and economic, the savings potential in the California market for rooftop air conditioning is 8.5 billion kWh in energy and \$1 billion per year<sup>1</sup>. This corresponds to a reduction in electric demand in California of 5,500

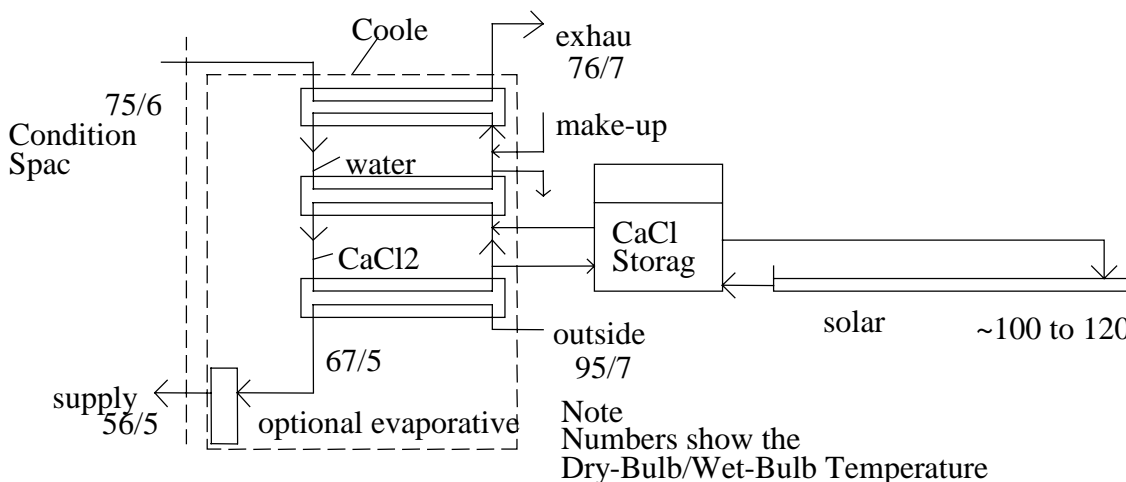
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1 J. Shugars, P. Coleman, C. Payne, and L. Van Wie McGrory. 2000. Bridging the Efficiency Gap: Commercial Packaged Rooftop Air Conditioners. In *Proceedings, 2000 ACEEE Summer Study*. LBL-46190, p. 10.217.

California market was estimated at 15% of U.S. market. Potential savings was estimated at 75% of rooftop air conditioning energy consumption. An average electricity price of \$0.12 per kWh for commercial customers was assumed.

MW<sup>2</sup>. The potential NO<sub>x</sub> reduction in California is 2,400 tons annually, based on the state average 2000 annual NO<sub>x</sub> output emission rate of 0.564 lb/MWh<sup>3</sup>.

The research concept couples modest-cost, low-temperature collectors with a low-cost calcium chloride solution for desiccant dehumidification and thermal storage. The addition of an evaporative cooler produces air conditioning at a competitive cost. The economic viability of this concept depends on optimizing the system and its components and on developing two key innovations – a low-cost heat exchanger and a solar, thermal, desiccant-regeneration subsystem. A process-flow diagram for the solar air conditioner appears below.



A calcium chloride solution concentrates (regenerates) while passing over the solar collector array. The concentrated calcium chloride solution cools via a plastic liquid-to-liquid heat exchanger and is then exposed to incoming outside air through a direct-contact enthalpy exchanger, similar to an evaporative cooler. Exhaust air from the conditioned space cools evaporatively, and the cooled water from the evaporative cooler sump reduces the temperature of the warm, concentrated calcium chloride solution through the plastic heat exchanger.

### 1.1. Objectives

The goal of this project was to determine the feasibility of a thermally driven, solar air-conditioning system employing desiccant dehumidification, low-temperature collectors for

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2 Based on an average rooftop air conditioning demand of 1500 full load hours per year

3 U.S. Environmental Protection Agency Emissions & Generation Resource Integrated Database (eGRID)

desiccant regeneration, and evaporative cooling. The researchers established the following project objectives:

1. Identify a design for a leak-free, plastic, liquid-to-liquid heat exchanger with a heat-transfer coefficient of 50 BTU/hr/ft<sup>2</sup>/°F.
2. Achieve collector water-evaporation rate of 1.0 lbm/day/ft<sup>2</sup>.
3. Plan for the first cost in the same range as high-efficiency electric A/C evaporator rooftop systems (\$1,800 to \$2,200 per ton installed).

## **1.2. Outcomes**

1. Nine different designs of low-cost, liquid-to-liquid heat exchangers were evaluated; two were tested. The preferred configuration consisted of plastic sheets welded together to form two counter-flow channels. It was the easiest to assemble and achieved a heat-transfer coefficient of 40 BTU/hr/ft<sup>2</sup>/°F, but the development of small, circuit-to-circuit leaks prevented lengthy and repeatable testing.
2. Sample solar-collector tests measured the evaporation rate from a calcium chloride solution. The sample collector consisted of a plastic plate filled with calcium chloride solution and covered with a black polyethylene film. The measured evaporation rates ranged between 0.5 and 1.0 lbm/day/ft<sup>2</sup> under partly cloudy, summer conditions in Northern Virginia.
3. The estimated cost per ton was based on modeled subsystem sizes and projected costs for materials, factory labor, mark-up, freight, and installation. The total projected price to an end user was \$1,825 per ton. Electricity requirements are expected to be on the order of 0.25 kW/ton, about ¼ that of a high-efficiency electric rooftop package. Annual water usage is estimated at 6,000 gallons per ton for a typical California application.

## **1.3. Conclusions**

1. Low-temperature solar collectors can provide effective regeneration of a calcium chloride liquid desiccant solution. Average evaporation rates of 0.5 to 1.0 lbm/ft<sup>2</sup>/day are achievable in California with simple open-collector designs.
2. A low-cost, high-performance, liquid-to-liquid heat exchanger was tested with an overall heat-transfer coefficient in the range of 40 BTU/hr/ft<sup>2</sup>/°F. Leakproof construction and longevity are important areas for future attention.
3. A projected 75% reduction in electricity use for air conditioning corresponds to an electric COP equivalent of 14. By contrast, a high-efficiency rooftop package has a COP of 3.5.
4. Water requirements appear to be modest and do not add appreciably to the operating expenses. At \$3/1,000 gallons, the cost for water is \$18/ton/yr, a fraction of the cost for an electric A/C evaporator unit. If the entire California inventory of rooftop air conditioning switched to this approach, the annual water requirements would be



100,000 acre-feet. In contrast, California consumes 9.5 million acre-feet annually for urban uses<sup>4</sup>.

5. Modeling, coupled with preliminary cost estimates for materials, labor, markups, and installation, indicates that the solar air-conditioning system has the potential to achieve installed costs of \$2,000/ton, on par with the typical installed cost of efficient rooftop models.

The technical feasibility of a novel solar air conditioner incorporating low-cost materials has been proved. Simple heat and mass-transfer tests were performed with representative material samples. The measured properties were used to size and cost the system. Beyond the scope of this small grant, considerable work remains to scale up the subsystems; to test prototype systems for performance and durability in an outdoor environment; and to confirm cost estimates for manufacturing, distribution, and installation.

#### **1.4. Recommendations**

After taking into consideration (a) research findings in the grant project, (b) overall development status, and (c) relevance of the technology to California and the PIER program, the Program Administrator has determined that the proposed technology should be considered for follow-on funding within the PIER program.

Receiving follow-on funding ultimately depends upon (a) availability of funds, (b) submission of a proposal in response to an invitation or solicitation, and (c) successful evaluation of the proposal.

The solar air-conditioning concept is a novel approach that recognizes the importance of initial cost to economic viability and market acceptance. Although the scale-up of the concept, its durability, and its true cost remain uncertain, it merits funding for the next development step. Further work should address the following:

- Heat-exchanger design and fabrication techniques for low cost and high performance.
- Heat-exchanger material that is inexpensive yet durable for ten to fifteen years of operation.
- Collector subsystem design that meets the \$70/ton material cost target yet is rugged enough to endure ten years of outdoor operation.
- Bench-scale system test to verify cooling capacity and parasitic electricity requirements.
- Following additional research and laboratory prototype testing, verification of the \$2,000-per-ton installed target requires in-depth analysis of material, manufacturing, distribution, sales, and installation costs.

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<sup>4</sup> California Department of Water Resources

## **1.5. Benefits to California**

Public benefits derived from PIER research and development are assessed within the following context:

- Reduced environmental impacts of the California electricity supply or transmission or distribution system
- Increased public safety of the California electricity system
- Increased reliability of the California electricity system
- Increased affordability of electricity in California

The primary benefit to the ratepayer from this research is increased affordability of electricity in California. The novel solar-air-conditioning concept would reduce the biggest cause of peak electricity demand. That would enable increased utilization of the generation, transmission, and distribution system and would delay new generating and transmission investments, lowering the cost of delivered electricity. Reducing peak demand also helps relieve congestion and improves the reliability of the power supply.

An economic solar air conditioner would also help California adopters of the technology control their energy expenses. The light commercial and small industrial sectors would best be able to utilize this technology. The electricity usage for rooftop air conditioning in California is 11.4 billion kWh per year<sup>5</sup>. With energy savings of 75% projected, the displacement potential of solar air conditioning in these California sectors is estimated at 8.5 billion kWh annually. That corresponds to a demand reduction in the vicinity of 5.5 GW and a consumer cost savings of \$1 billion per year.

## **1.6. Overall Technology Transition Assessment**

As the basis for this assessment, the Program Administrator reviewed the researcher's overall development effort, which includes all activities related to a coordinated development effort, not just the work performed with EISG grant funds.

### **Marketing/Connection to the Market**

The investigator reports that discussions with building developers and owners indicate a high degree of interest. Preliminary cost analysis suggests that a solar system would compete in new and replacement rooftop air-conditioning applications, creating a very large potential market. However, the market is cost sensitive first of all, and further work is required to shore up performance and cost projections.

### **Engineering/Technical**

The project showed that the basic approach is technically feasible. Preliminary sizing and costing of the key components and subsystems suggests that the low-cost target is reachable. A

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<sup>5</sup> See footnote # 1

bench-scale prototype and additional heat-exchanger design development will help better define collector and heat-exchanger sizing and costs.

### **Legal/Contractual**

A patent on the solar-air-conditioning concept has been granted. Patent number is US 6,513,339

### **Environmental, Safety, Risk Assessments/ Quality Plans**

This project is in an early stage of research and development. It is premature to conduct environmental, safety, or risk assessments, or to develop Quality Plans. There are relevant issues that need to be considered during advanced development stages, including heat-exchanger performance and durability, calcium chloride carry-over, and maintenance requirements.

### **Production Readiness/Commercialization**

This project is in early phases of research and development and not yet ready for commercialization planning. WorkSmart should consider teaming with a plastics manufacturer for follow-on development work.

Appendix A: Final Report (under separate cover)

Appendix B: Awardee Rebuttal to Independent Assessment (none submitted)

**ENERGY INNOVATIONS SMALL GRANT  
(EISG) PROGRAM**

**EISG FINAL REPORT**

**Title: Evaluation of a New Solar Air Conditioner**

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Inquires related to this final report should be directed to the Awardee (see contact information on cover page) or the EISG Program Administrator at (619) 594-1049 or email [eisgp@energy.state.ca.us](mailto:eisgp@energy.state.ca.us).

## Table of Contents

<b>Abstract.....</b>	<b>1</b>
<b>Executive Summary.....</b>	<b>1</b>
<b>Introduction.....</b>	<b>3</b>
<b>Project Objectives.....</b>	<b>8</b>
<b>Project Approach.....</b>	<b>9</b>
<b>Project Outcomes.....</b>	<b>9</b>
<b>Conclusions.....</b>	<b>23</b>
<b>Recommendations.....</b>	<b>23</b>
<b>Public Benefits to California .....</b>	<b>23</b>
<b>Development Stage Assessment .....</b>	<b>24</b>
<b>References.....</b>	<b>25</b>

**Appendix: US Patent 6,513,339**

## **List of Figures**

1. Average Global Horizontal Radiation for California in August
2. Schematic Diagram of the Solar Air Conditioner
3. Evaporation Test Data from Two Open-Pool Solar Collectors
4. Weather Data During Collector Tests
5. First Liquid Heat Exchanger Geometry
6. Second Liquid Heat Exchanger
7. Comparison Between Curve-Fit Effectiveness and Manufacturer's Data for an Evaporative Pad
8. Comparison Between Curve Fit and Manufacturer's Pressure Drop Data for an Evaporative Pad
9. Modeled Cooler Temperatures
10. Effect of Outside Air Requirement on Rated Capacity for Typical California Design Conditions
11. Development Assessment Matrix

## **List of Tables**

1. Cooler Model Results
2. Projected Costs for New Solar Air Conditioner

## Abstract

Components for a new, patented, thermally driven, solar air-conditioning system were tested and analysis of the performance of the system was evaluated.

The proposed system uses an inexpensive desiccant, such as calcium chloride, in combination with an indirect evaporative cooler to provide cooling and dehumidification. Calcium chloride absorbs moisture from incoming air. Moisture from the calcium chloride is evaporated in a low-cost solar collector that operates at a very low temperature (typically less than 120°F). The collector can be a simple pool in a black plastic liner that is open to outside air. A transparent cover may be added to prevent accumulation of rainwater. Concentrated calcium chloride solution can be stored for use at night or during cloudy periods when sunlight is not available. Evaporation rates of approximately .5 to 1.0 lbm/ft<sup>2</sup>/day were demonstrated in a small pool under summer conditions.

The heat-exchange system uses 100% outside air. An indirect evaporative cooler uses exhaust air to cool desiccant liquid, which cools incoming air. Modeling shows a coefficient of performance (COP) of 1.35 assuming no credit for outside air. If ventilation load is included (100% outside air requirement), the COP is approximately twice this value. Projected peak electrical demand (mainly fans and pumps) is about 25% of a high-efficiency base rooftop unit. Estimated factory cost of the system is approximately \$430 per ton with an installed cost of less than \$2000/ton for rooftop applications. These projected costs should be competitive with conventional rooftop air-conditioners on a first-cost basis.

## Executive Summary

**Introduction:** Solar air conditioning has great potential. Sunlight is most plentiful in the summer when cooling loads are highest. For a typical single-story building in California, the daily amount of solar energy available on the roof is roughly 10 times the daily cooling load.

Unfortunately, currently available technologies have not been able to economically use this huge resource. For example, photovoltaic (PV) cells are expensive and require a large portion of the roof to provide sufficient electrical power to drive an air conditioner. The cost for these panels can be ten times that of a conventional air conditioner.

Energy storage is another problem with existing systems. Maximum solar energy generally occurs around solar noon, while the maximum cooling load is several hours later. Storing electricity, such as with batteries, is costly and introduces additional energy losses. Storing cooling in the form of ice or cold water takes a large amount of space in addition to the problems with cost and efficiency. Of course the electrical grid can be used to supplement the energy from the PV panels, but this approach means that the generation and transmission capacity must be available to handle the maximum cooling load, which negates much of the benefits from solar air conditioning.

Thermally driven systems have also seen some use, but have similar problems. These systems use conventional absorption chillers or other cooling equipment that is designed for use with natural gas or steam as an energy source. Unfortunately, the high temperatures required to drive these systems greatly increase the cost of the solar collectors. In addition there is no economical way of storing the thermal energy.

**New System:** The new solar air-conditioning system takes a much different approach. It is a thermally driven system that differs from earlier thermal systems in that it is designed to work with low-temperature solar collectors and includes low-cost energy storage. Energy is stored in the form of a concentrated salt (desiccant) solution that can be used to provide dehumidification. A cooler with a special heat exchanger design combines the dehumidification from the desiccant with cooling from evaporation of water to provide air conditioning.

The preferred salt is calcium chloride. It is inexpensive and is commonly used as road salt. The solar collector can be a shallow pool in a black plastic liner, which allows the sun to evaporate water from the salt solution. The concentrated salt solution is then stored in a tank until it is needed for cooling. If sunlight is not sufficient to concentrate the desiccant, off-peak electricity or natural gas can be used as a back-up heat source. (For a detailed technical description of the solar air conditioner, see pages 6 and 7 in the Introduction.)

**Project Objectives:** The overall objective of the project is to demonstrate a thermally driven solar air conditioner that has the potential of being economically viable compared to conventional electrically driven systems. The immediate objectives were to obtain component test data and modeling results that can be used to support the overall objective.



**Project Outcomes and Conclusions:** There were several important outcomes from this work:

1. Low-cost solar collectors can provide effective evaporation of water from the desiccant liquid.
2. Several low-cost, high-performance liquid-to-liquid heat exchangers used in the cooler were tested and have promising performance. Leak-proof construction needs further work.
3. A 75% reduction in peak electrical demand from air conditioning is feasible based on modeling results.
4. Modeling shows that solar air-conditioning systems have the potential to be competitive on a first-cost basis with conventional air conditioners.

**Recommendations:** This work shows a great potential for solar air conditioners. Further work on the development of the cooler heat-transfer is fundamentally important. The use of low-cost plastic materials that eliminate corrosion problems with conventional metal heat exchangers offers great promise for achieving low-cost, high-performance designs that are essential for creating an economically viable solar air conditioner.

## **Introduction**

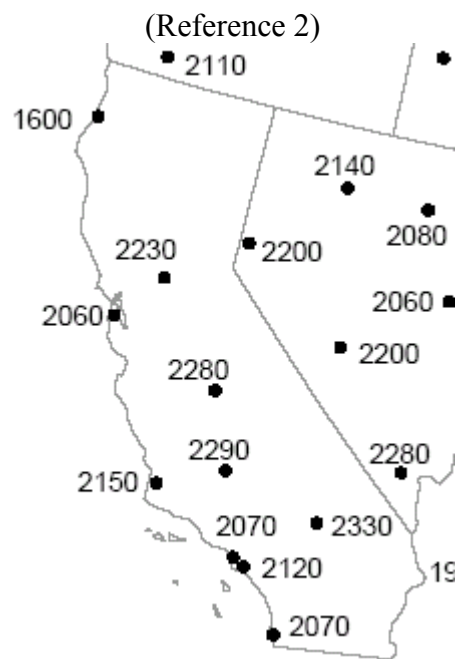
Electrically driven air conditioning uses large amounts of electrical energy and greatly increases the peak electrical power demand for power companies who supply this electricity. In California, supplying this electricity costs billions of dollars each year for fuel. Additional billions are required for building and maintaining the electrical generating facilities and transmission systems necessary to supply this electricity.

From an environmental standpoint the energy used by air-conditioning systems is especially undesirable. In order to meet the high electrical demand created by air conditioning, power companies need to operate older, less-efficient power plants that usually have much higher emission rates than more-modern equipment that is used to handle base-load demand. In addition, weather conditions that create high temperatures that correspond peak air-conditioning

loads frequently also create conditions that are favorable for the formation of photochemical smog. Continued rapid development of interior regions of California with high summer temperatures and large air-conditioning requirements further contributes to these problems.

On the surface, air conditioning is an ideal application for solar energy; sunlight is especially plentiful during times of the year when air condition loads are typically the highest. The map below shows the average amount of solar energy hitting a horizontal surface for California in August. The average solar radiation throughout California is approximately 2000 Btu/day/ft<sup>2</sup>. By comparison, the average cooling load per day is on the order of 200 Btu/day/ft<sup>2</sup> of floor area. These figures show that for a COP of ~1.0 the available solar energy hitting a roof is factor of ten greater than the energy required for cooling a single-story building.

Figure 1: Average Global Horizontal Solar Radiation for California in August (Btu/ft<sup>2</sup>/day)



*The problem is that existing solar technologies have not produced systems that are economically competitive with conventional electrically driven systems. Prior work with solar air conditioning has not produced practical systems.*

Solar air conditioning systems have used two basic approaches in an attempt to capture the sun's energy for cooling—thermal and photovoltaic (1). The photovoltaic systems use

photovoltaic panels to convert solar radiation directly into DC electricity. Photovoltaic systems have two major advantageous attributes. First, they can use conventional electrically driven air-conditioning equipment, which is widely available and inexpensive. Second, they can use the utility grid for backup power during dark or cloudy periods.

Unfortunately other attributes: the high cost of manufacturing, the low conversion efficiencies, and the need for a continual stream of photons to produce power, create three major disadvantages. First electricity from solar cells is very expensive because of the high cost of the solar panels. (Panels for a residential air conditioner can cost tens of thousands of dollars.) Second the space needed for powering the air conditioning units is large. And third the panels provide no energy storage, which creates a need for use of grid-based electricity at night and on cloudy days. In fact, the peak output from the solar panels occurs around solar noon, while peak air-conditioning loads occurs several hours later, resulting in a significant mismatch between supply of needed power and demand. This mismatch greatly reduces the value of the system in reducing peak power demand to the utility. Recently deregulated markets are demonstrating that these demands are much more expensive to meet than had been previously apparent.

For off-grid locations, the only viable energy storage system to match the provision of power to times when demand is high (later in afternoon and at night) is batteries. Batteries have a high first cost, require periodic replacement, and normally use toxic and/or corrosive materials. These problems have prevented the use of photovoltaic systems in other than a few high-cost demonstration systems.

Thermally driven systems are another approach; they use heat from the sun to drive an air conditioner. Typical approaches from the past used a high-temperature flat-plate collector to supply heat to an absorption system. Systems with concentrating collectors and steam turbines have also been proposed. Natural gas or other fuel is used for backup heat.

While thermal systems have the advantage of eliminating the need for expensive photovoltaic panels, the existing systems have attributes that produce major disadvantages. As used in the past, thermal systems are based on single-effect absorption chillers or other cooling systems that are designed to use natural gas, steam or other high-temperature heat source. They require a very high collector temperature ( $\sim 180^\circ\text{F}$ ) to drive the cooling system. The high collector temperature and relatively poor efficiency ( $\text{COP} = \sim .6$ ) greatly increases collector size and cost. In addition, there is no economically viable way of storing solar energy with this

approach. The result of these problems is that thermal systems have been very expensive and have relied primarily on natural gas or other fuel for their thermal energy. For this reason they have seen very little use.

**Detailed Description of the Proposed Solar Air Conditioner:** Figure 2 is a schematic diagram of a proposed thermally driven solar air conditioner that addresses the problems of the earlier systems. The basic idea is to use a desiccant liquid (preferably calcium chloride,  $\text{CaCl}_2$ ) for cooling and dehumidifying. The cooler uses the calcium chloride to dehumidify the air. Water is evaporatively cooled with exhaust air. Approximate air dry-bulb/wet-bulb temperatures in degrees F are included in the figure for illustration purposes. This schematic shows the configuration of cooler evaluated in this project; other configurations are possible and are discussed in the copy of the patent in the appendix.

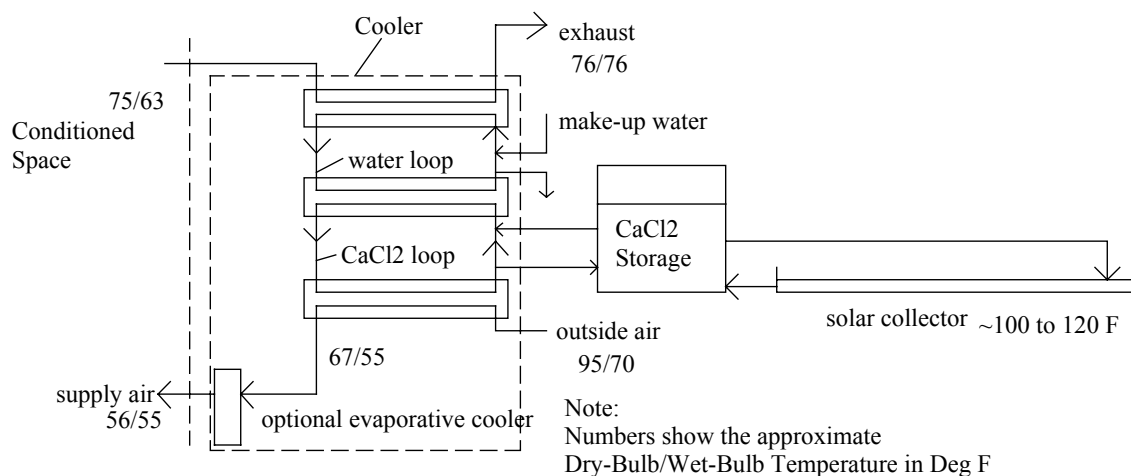


Figure 2: Schematic Diagram of the Solar Air Conditioner

The operation of cooler requires further explanation. The driving force behind this cooler is a temperature difference between a water surface and a desiccant surface in contact with an air stream. The temperature of a water surface is close to the wet-bulb temperature of the air, while the temperature of a desiccant surface corresponds to a higher temperature. For a concentrated solution of calcium chloride, the temperature difference between the wet-bulb and the desiccant equilibrium temperature is approximately 15 to 20 °F. The cooler provides a way of using a relatively weak desiccant to efficiently provide cooling.

The cooler uses three counterflow heat exchangers to move thermal energy from incoming outside air to the exhaust air. Starting at the top of the figure, the first heat exchanger

is a direct-contact device that evaporates water into the exhaust air stream. The temperature of water leaving this heat exchanger approaches the wet-bulb temperature of the exhaust stream leaving the conditioned space.

The second heat exchanger is a liquid-to-liquid heat exchanger that uses the cooled water to cool a calcium chloride solution. The third heat exchanger is a direct-contact device that cools and dehumidifies the incoming outside air using the cooled calcium chloride solution.

The cooler is able to work because the equilibrium temperature of the desiccant with the incoming air is higher than the wet-bulb temperature of the exhaust air stream. This temperature difference allows the cooler to lower the enthalpy of the incoming air below that of the exhaust air leaving the occupied space. An optional direct evaporative cooler can provide a low supply-air temperature for applications where the cooling load is primarily sensible.

A solar collector uses thermal energy from the sun to drive out water absorbed by the desiccant. The collector temperatures are relatively low because the equilibrium relative humidity of the desiccant is relatively high. For example, for an ambient wet-bulb temperature of 70°F and 40% equilibrium relative humidity, water will start to evaporate from the desiccant at a temperature of only 89°F. Typical operating collector temperatures in the range of 100°F to 120°F are possible. Note that if the ambient temperature is above the equilibrium temperature for the desiccant it is possible to evaporate water into the air without any solar input.

The concentrated desiccant solution also serves as a storage medium to provide cooling at night or during cloudy periods, when sufficient sunlight is not available. The preferred desiccant material is calcium chloride because of its low cost and low toxicity. (It is commonly used for road salt and as a food additive.) Because a desiccant uses the heat of vaporization of water to store energy, the possible energy storage density for cooling is approximately 300-500 Btu/lbm of salt (~200 Btu/lbm of solution). This energy density compares favorably to the energy density of ice, which is roughly 150 Btu/lbm, and chilled water, which is roughly 20 Btu/lbm. The retail price of calcium chloride is less than \$.25/lbm. These features mean that calcium chloride can serve as a safe, compact, inexpensive storage medium.

On the other hand, the desiccant properties of calcium chloride impose some limitations. At 25 °C (77 °F) calcium chloride solution crystallizes at a concentration above about 50%. This concentration corresponds to a relative humidity of about 30%; operation of the system requires that the solution be kept safely above this concentration. In contrast, more-conventional

desiccants, such as lithium chloride, crystallize at much higher concentrations and can achieve a much lower relative humidity. Unfortunately, lithium chloride costs over 20 times as much as calcium chloride and has some toxicity issues, which makes it undesirable for use as a thermal storage medium. Designing for a higher relative humidity that is available with calcium chloride also allows for much lower solar collector temperatures, which can greatly reduce collector cost.

An important feature for making the system work is to develop very low-cost, high-performance heat exchangers. In order to make calcium chloride work effectively, the heat exchangers must handle large transfer large amounts of thermal energy at a small temperature difference.

As will be discussed in more detail later, the design approach used in this project was make use of low-cost materials, such as plastic, as much as possible in the design of the heat exchangers. Plastics have the advantage of being resistant to the corrosive effects of salt solutions and water. The disadvantage is that plastic has a much lower thermal conductivity and lower strength compared to metals. The design approach was to use large areas of primary surface with very small pressure differences.

## **Project Objectives**

The overall objective of the project is to demonstrate a thermally driven solar air conditioner that has the potential of being economically viable compared to conventional electrically driven systems. For widespread acceptance, this objective means having a first cost that is similar to or better than a base system. This objective is highly aggressive in that existing PV systems can cost a factor of ten more than conventional air conditioning systems.

The immediate objectives were:

1. Test performance of a low-cost solar collector,
2. Design and test low-cost heat-exchanger components, and
3. Model system performance and cost using test results.

## **Project Approach**

The project approach is driven toward achieving the ultimate objective of producing an economically viable solar air-conditioning system. This approach combined testing, modeling,

and design revisions that move toward achieving this overall objective. Test results were incorporated into a computer model to show the projected performance of the system.

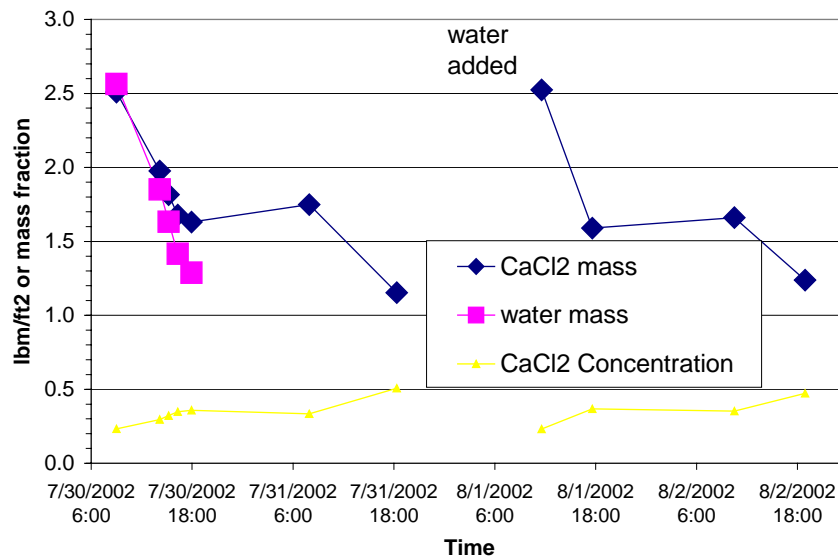
## **Project Outcomes**

**Solar collector:** Simple tests were performed to determine the evaporation rate of desiccant liquid from small pool collectors. Results of these tests are shown in figure 3. Figure 4 shows the dry-bulb temperature, dewpoint temperature, and relative humidity during the tests as measured by the National Weather Service at Regan National Airport, about 10 miles from the test location in Springfield, Virginia.

The test collectors were round plastic plates covered with 4-mil black polyethylene that were filled with a shallow pool of liquid. The surface area of the pool was approximately 6.5 inches (165 mm) in diameter. Most of the test data is from one collector that was filled with calcium chloride solution. A second collector was filled with water for one day. A balance provided a measurement of mass of the collector at various times throughout the day. Calcium chloride concentration was determined by measured liquid density at the end of the first day and was calculated based on the mass of the liquid at the other points. Air flow over the collector was by natural convection; wind speeds were generally less than 10 mph. At the beginning of the third day of measurements, a measured quantity of distilled water was added to the calcium chloride solution to prevent crystallization.

The measured evaporation rate corresponds to approximately .5 to 1.0 lbm/day/ft<sup>2</sup> under partly cloudy, summer conditions in northern Virginia. As expected the evaporation rate depended on the concentration of calcium chloride; higher concentrations reduce evaporation rate. Overnight the mass of calcium chloride solution increased, which reflects the absorption of moisture from the air during periods of high ambient relative humidity.

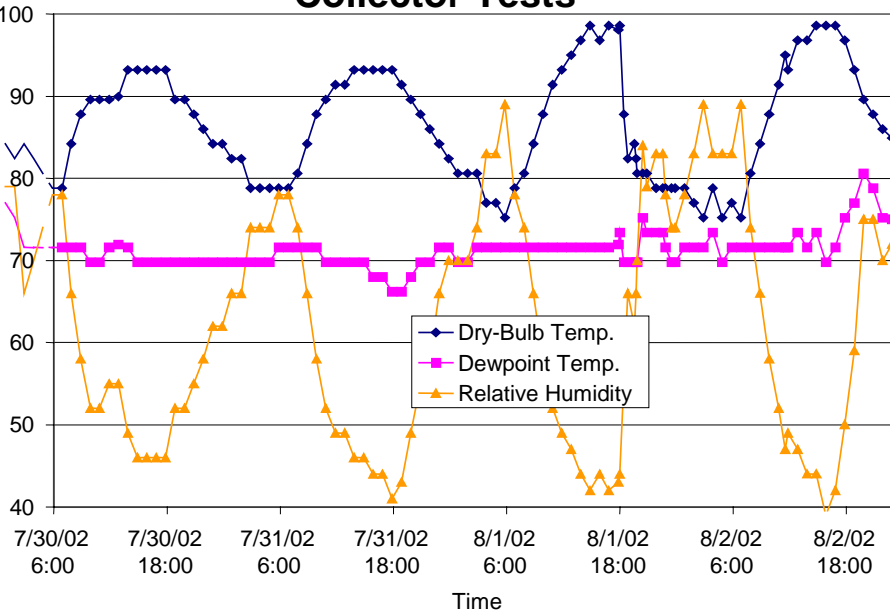
**Figure 3: Evaporation Test Data from Two Open-Pool Solar Collectors**



**Figure 4: Weather Data During Collector Tests**

Temperature in  
Deg. F or

Percent  
Relative  
Humidity



The actual performance of a collector in California should be significantly better than the test results from northern Virginia. Regions of California that require air conditioning generally have climates that are drier and sunnier than the climate of northern Virginia where the tests were performed. Lower ambient relative humidity should significantly increase the evaporation rates from open collectors. Indeed, if ambient humidity is sufficiently low, it is possible to evaporate



water from the desiccant solution without any sunlight. In addition, the intensity of solar radiation in most of California is significantly higher than that found near Washington DC.

**Heat-exchanger design and testing:** Several different designs were evaluated for a low-cost liquid-to-liquid heat exchanger.

Figure 5 shows one configuration of heat exchanger. The exchanger uses separate plastic bags for each fluid. The bags are constructed of polyethylene or other plastic. Spacers were used inside bags to create a thin channel, typically about .010 to .050 inches.

Several tests were made using variations of this design. The basic idea is to keep the liquid layers thin so that thermal resistance is small even with low-velocity laminar flow. The theoretical heat transfer coefficient should be about 100 Btu/hr/ft<sup>2</sup>/F. The measured heat transfer performance was variable with an overall heat-transfer coefficient of between 20 and 40 Btu/hr/ft<sup>2</sup>/F. Sealing the heat exchanger and maintaining the proper distribution of liquid through the heat exchanger was difficult. Air gaps between the bags can also introduce significant thermal resistance.

Figure 5: First Liquid Heat Exchanger Geometry

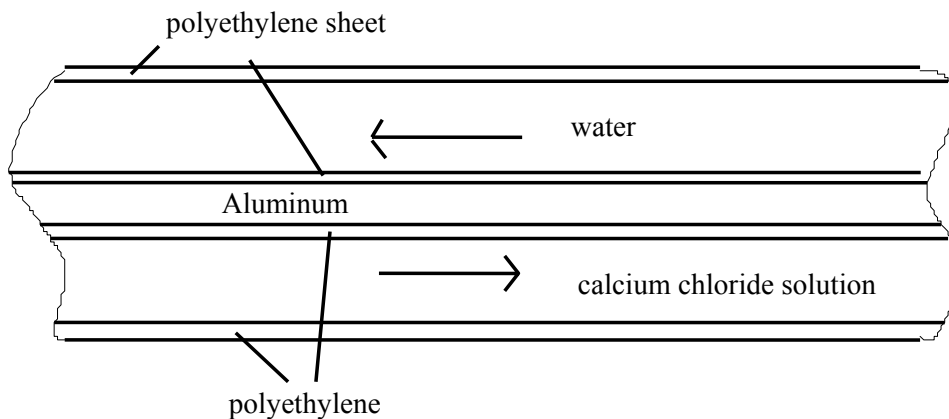
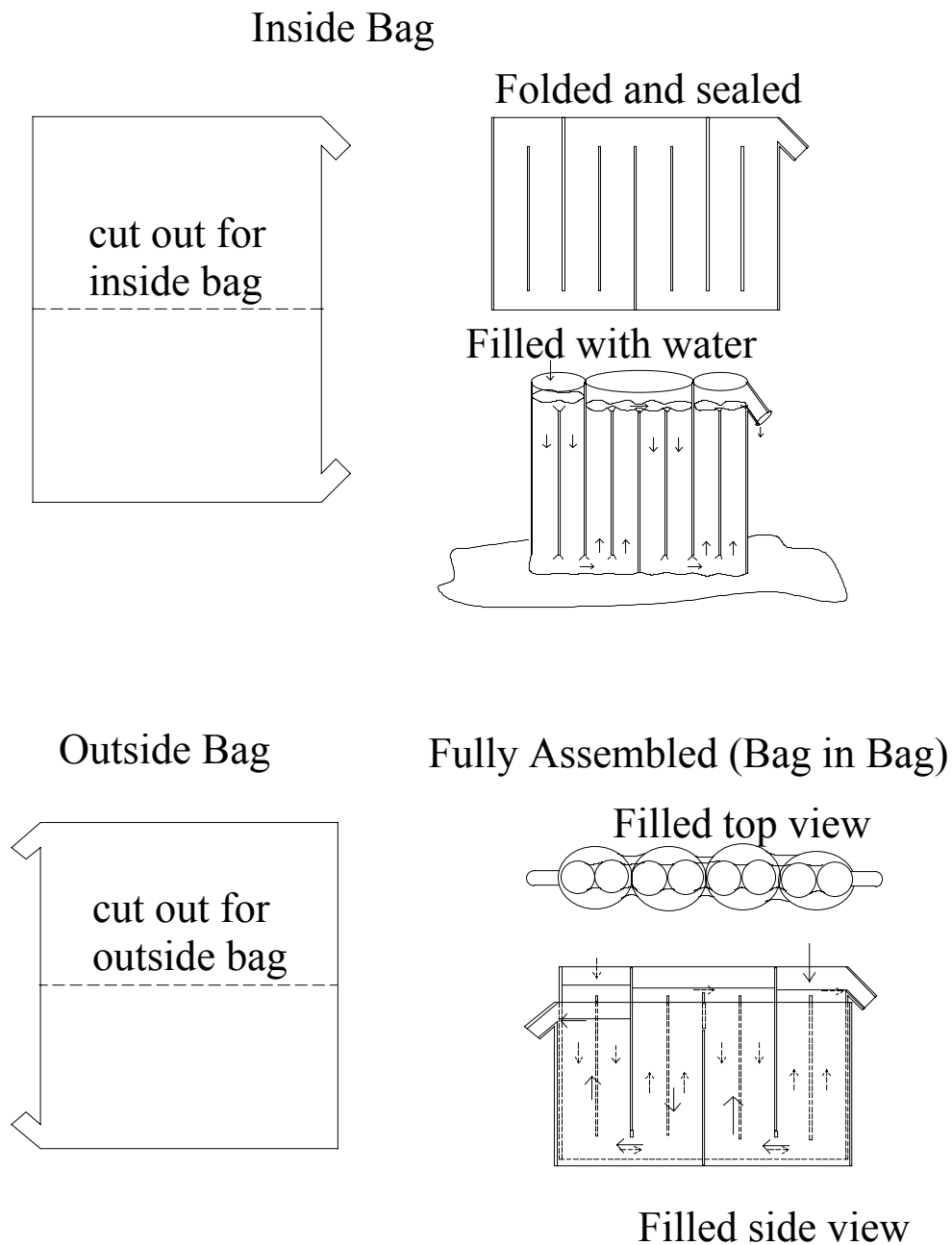


Figure 6 shows an alternate configuration of heat exchanger. This design is formed with flexible plastic sheets that are welded together to form two, counter-flow channels. The heat exchanger is open to the atmosphere at the top so that the only pressure is the static head of the liquids that fill the heat exchanger. This heat exchanger proved to be much easier to assemble and achieved a heat transfer coefficient of approximately 40 Btu/hr/ft<sup>2</sup>/F. Unfortunately a small circuit-to-circuit leak developed and testing and time did not allow for further testing of the system.

Figure 6: Second Liquid Heat Exchanger



Heat exchanger testing was conducted using a simple, once-through test apparatus. Warm city water was supplied to one side of the heat exchanger, and cool water was supplied to the other side. The flow rate was determined by measuring the time for water to fill a container and then weighing the mass of water collected using a balance. Temperature measurements used thermister probes with a digital readout. Calibration of the probes was checked in an ice bath.

Insulation was provided to the heat exchanger to limit convection loss to the surrounding air. Heat balances of 5 to 10% were achieved with this setup, but getting consistent, leak-free operation was a serious problem with the heat-exchanger tests.

**Modeling:** A computer model combined manufacturer's data for an evaporative pad was combined with measured performance data for the liquid heat exchanger and solar collector to develop performance and cost estimates for a complete solar air conditioner.

The first step was to model the liquid-to-air heat and mass exchanger. Figures 7 and 8 show that the curve fits agree closely with the manufacturer's data. The starting point for this analysis was a standard evaporative pad. The performance thermal performance of the pad was curve fit with the equation:

$$UA = (B v + C) V$$

UA = the available heat transfer coefficient in Btu/hr/F

B = a constant = 1.985

C = a constant = 245.1

v = air face velocity in ft/min

V = volume of heat evaporative pad in ft<sup>3</sup>.

Effectiveness for an evaporative cooler is then given by the equation:

$$\text{Eff} = 1 - \exp(-NTU)$$

where

Eff = effectiveness

NTU = number of heat transfer units = UA / C<sub>air</sub>

C<sub>air</sub> is the thermal mass flow rate of air through the evaporative cooler

$$C_{\text{air}} = \rho U c_p$$

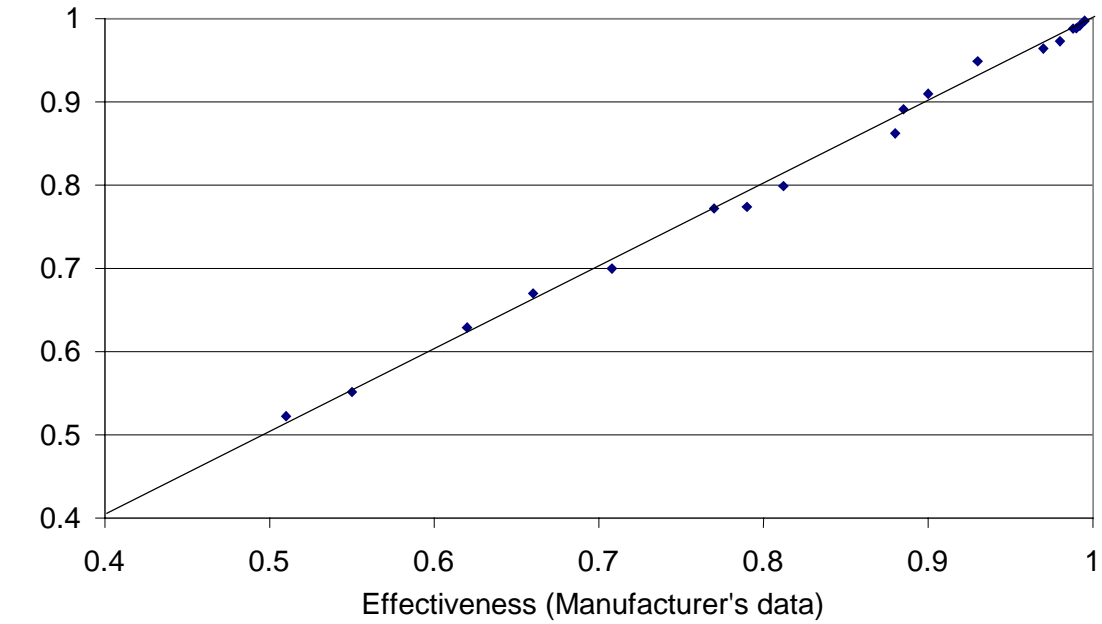
where

ρ = density of air in lbm/ft<sup>3</sup>

U = volumetric flow rate (ft<sup>3</sup>/sec)

c<sub>p</sub> = specific heat of air at constant pressure (.24 Btu/lbm/F)

**Figure 7: Comparison Between Curve-Fit Effectiveness and Manufacturer's Data for an Evaporative Pad**



Pressure drop curve fit is

$$\Delta p = (D v^2) L$$

where

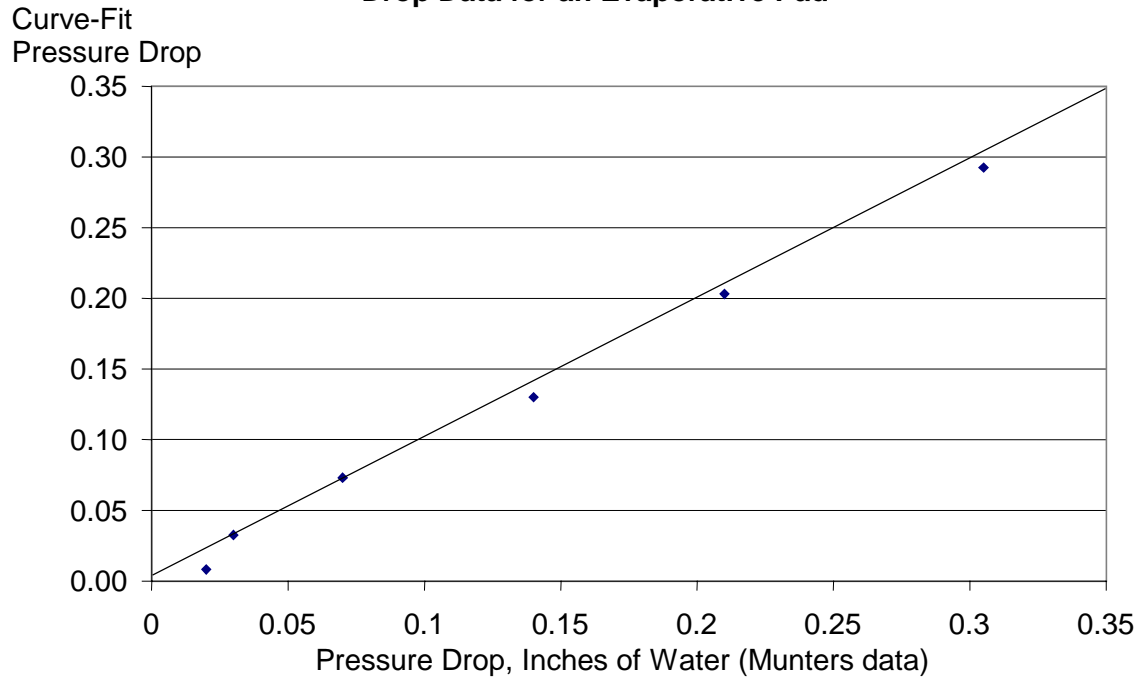
$\Delta p$  = pressure drop inches of water,

$D$  = constant =  $8.125 \times 10^{-7}$ ,

$v$  = air face velocity in ft/min, and

$L$  = depth of the pad in feet.

**Figure 8: Comparison Between Curve Fit and Manufacturer's Pressure Drop Data for an Evaporative Pad**



The analysis of the heat and mass transfer uses an extension to the analogy based on enthalpy:

- 1) Enthalpy change ( $\Delta h$ ) is analogous to temperature change ( $\Delta T$ ) and
- 2) Mass flow rate ( $m$ ) is analogous to thermal mass flow rate ( $c$ ). (See reference 3 for background on enthalpy analogy.)

This analogy allows the use of conventional heat exchanger relations, which greatly simplifies the analysis of the heat and mass transfer. Enthalpy is evaluated for the airside and is determined by the equilibrium conditions at the surface. For a surface that is wetted with water, the equilibrium enthalpy corresponds to the enthalpy of saturated air at the surface temperature. For a surface that is wet with desiccant, the equilibrium enthalpy of air corresponds to air at the equilibrium relative humidity of the liquid and the surface temperature.

For the airside  $m$  is simply the mass flow rate of dry air. For the liquid side of a heat exchanger,  $m$  is the mass flow rate of air that would give the same enthalpy change with the same temperature change. Since the enthalpy of the air depends on the temperature, it may be necessary to iterate to find the equivalent air mass flow rate. This approach allows the use of standard NTU-effectiveness relations in calculating the performance of devices with simultaneous heat and mass transfer.

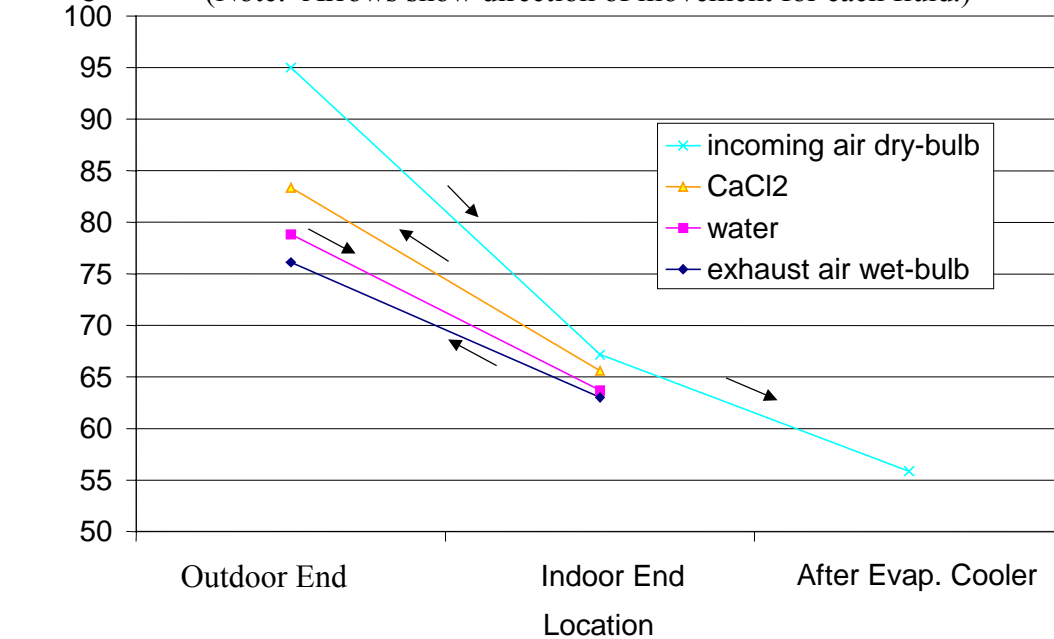
**Modeling results:** Figure 9 and Table 1 show the modeled results. The model is an EXCEL spreadsheet that uses Visual Basic macro routines for calculating psychrometric and heat-exchanger functions. The projected thermal COP for the system is 1.36 for a typical summer design condition in California (95 °F dry-bulb/70 °F wet-bulb). This COP is based on the enthalpy difference between the supply air and the room air entering the cooler. This enthalpy difference corresponds to an application where no credit is given for bringing in outside air.

Figure 10 show the effect of the outside air requirement on the rated COP and capacity of the system. If full credit is taken for the enthalpy difference between the supply air and the outside air, then the calculated cooling capacity and thermal COP is approximately doubled. This second case applies to cases where 100% outside air is required. For typical commercial applications that require 20% outside air, the rated capacity and COP is increased by about 20% to account for the extra cooling load for the base system.

(The solar air conditioner inherently uses 100% outside air so the question is how much credit should it receive for cooling this air. The enthalpy of return air with a 63 °F wet-bulb temperature is about 28.6 Btu/lbm, while outside air with 70 °F wet-bulb temperature has an enthalpy of about 34.1 Btu/lbm. For a supply air enthalpy of 23.2 Btu/lbm, the enthalpy change is roughly twice as large for cooling outside air as for cooling return air. A conventional air conditioner would thus have roughly twice the cooling load for cooling 100% outside air as for 100% return air. However, if the application does not require 100% outside air, then a fair evaluation of the solar air conditioner would only take credit for cooling the required quantity of outside air to the return-air conditions.)

If standard ARI conditions are considered (80 °F dry-bulb, 67 °F wet-bulb inside and 95 °F dry-bulb, 75 °F wet-bulb outside), the capacity of the system increases by about 9% and the COP declines to .86. This analysis shows that the system is capable of handling typical design conditions found in the eastern US, although there is a substantial penalty in the thermal efficiency. As a practical matter, installation in humid climates of the eastern US would require substantially larger collector area, but should otherwise function in an acceptable manner. This feature is important in commercializing the product since manufacturers are reluctant to develop a specialty product that is confined to only one region of the country.

**Figure 9: Modeled Cooler Temperatures**  
(Note: Arrows show direction of movement for each fluid.)



**Figure 10: Effect of Required Outside Air on Rated Capacity or COP for Typical California Design Conditions (95 F dry-bulb / 70 F wet-bulb)**

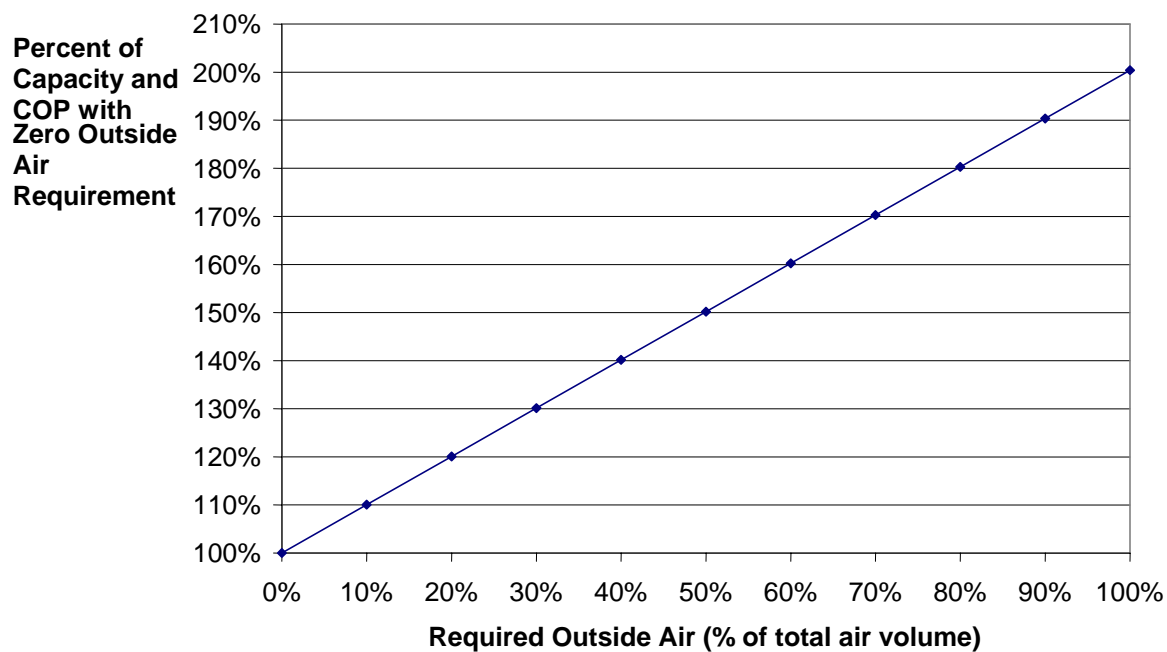


Table 1: Cooler Model Results

	exhaust hx	inlet air hx
(calculated) inlet liquid temperature	78.824	65.598 F
COP (recirc)		1.3591
Net cooling		12084.1695 Btu/hr
<b>INPUTS</b>		
liquid specific heat	1	0.65 Btu/lbm/F
liquid equilibrium relative humidity	100%	45%
liquid mass flow rate (lbm/hr)	1600	2100 lbm/hr
inlet air dry-bulb temperature	75	95
inlet air wet-bulb temperature	63	70 F
air volumetric flow rate	500	519.1858365 CFM
altitude	0	0 ft
heat exchanger UA (sensible value)	4500	4500 Btu/hr/F
face velocity (fpm)	500	519.19 ft/min
Evap hx slope	1.985	1.985 Btu/hr/F/ft3/ft m
Evap hx intercept	245.107	245.107 Btu/hr/F/ft3
liquid hx heat transfer coefficient	40	Btu/hr/F/ft2
liquid hx area	200	ft2
<b>CALCULATED RESULTS</b>		
<b>Liquid hx:</b>		
c	1600	1365 Btu/hr/F
cmin	1365	Btu/hr/F
cmax	1600	Btu/hr/F
cmin/cmax	0.853125	dimensionless
UA	8000	Btu/hr/F
NTU	5.860805861	dimensionless
effectiveness	0.902856649	dimensionless
actual effectiveness	0.902856649	dimensionless
heat transfer	24216.00869	Btu/hr
calculated leaving liquid temperature	78.82399648	65.5978076 F



Table 1 (Continued): Cooler Model Results

<b>Liquid-air heat exchanger calculations:</b>		
	<b>exhaust hx</b>	<b>inlet air hx</b>
airside UA (in terms of enthalpy difference)	18750	18750 Btu/hr/(Btu/lbm of dry air)
Evap. Media UA/ft <sup>3</sup>	1238	1276 Btu/hr/F/ft <sup>3</sup>
required hx volume	3.635536491	3.526999211 ft <sup>3</sup>
required hx face area	1	1 ft <sup>2</sup>
required hx depth	3.635536491	3.526999211 ft
pressure drop	0.73846835	0.772457163 inches of water
barometric pressure	14.7	14.7 psia
inlet air RH	52.15%	29.12%
inlet air w	0.009639768	0.01022755 lbm water/lbm dry air
air specific volume	13.68472635	14.20983219 ft <sup>3</sup> /lbm of dry air
air mass flow rate	2192.225057	2192.225057 lbm/hr
inlet air enthalpy	28.54879844	34.08282843 Btu/lbm dry air
inlet air equilibrium temperature	63.04631266	86.44514003 F
inlet water surface enthalpy	42.34271798	22.28633729 Btu/lbm dry air
<b>Third (final) iteration:</b>		
calculated equivalent air flow rate of liquid	1817.710863	2483.569799 lbm/hr
Cmin	1817.710863	2192.225057 lbm/hr
Cmax	2192.225057	2483.569799 lbm/hr
NTU	10.31517189	8.552953966 dimensionless
Effectiveness	0.965806386	0.936406846 dimensionless
heat transfer	24216.00869	-24216.00869 Btu/hr
liquid outlet surface enthalpy	29.0204624	32.03682173 Btu/lbm of dry air
liquid outlet temperature	63.68899105	83.33847331 F
liquid surface enthalpy	29.01929934	32.03616058 Btu/lbm of dry air
air outlet enthalpy	39.59511351	23.03651336 Btu/lbm of dry air
leaving air wet-bulb temperature	76.11086263	54.77237856 F
leaving air equilibrium temperature	76.11086263	67.16596366 F
approximate sensible effectiveness	0.999807026	0.999807026 dimensionless
approximate leaving air temperature	76.11064826	67.17133491 F
inlet air enthalpy change		11.04631507 Btu/lbm/hr
Gross cooling capacity (outside air credit)		24216.00869 Btu/hr
Enthalpy change (return - supply)		5.512285076 Btu/lbm of dry air
Net cooling capacity (recirculation)		12084.16947 Btu/hr
RH	100.00%	45.16%
Wout	0.019481624	0.006364826 lbm water/lbm dry air
Win	0.009639768	0.01022755 lbm water/lbm dry air
W change	0.009841855	-0.003862724 lbm water/lbm dry air

**Table 1 Continued**

water added	21.57556216	-8.467959291	lbm/hr
heat of vaporization	1050	1050	Btu/lbmF
Latent heat absorbed by desiccant		-8891.357256	Btu/hr
COP		1.35909	dimensionless

**Optional evaporative cooler**

	exhaust hx	inlet air hx
design effectiveness		90%
Velocity		400 ft/min
height		3 ft
water flow requirement		1.25 gpm/ft2
outlet dry bulb temperature		55.855 F
outlet wet bulb temperature		54.77237856 F
UA/ft3		1039 Btu/hr/F/ft3
NTU		2.302585093 dimensionless
cmin		526.1340138 Btu/hr/F
UA required		1211.468337 Btu/hr/F
ft3 required		1.165717802 ft3
face area		1.297964591 ft2
depth		0.898112175 ft
length		0.432654864 ft
pressure drop		0.116754583 inches of water
water flow requirement		0.485715751 gpm

## Projected Costs:

Table 2: Projected Costs for New Solar Air Conditioner

Cost analysis on a per ton with 20% outside air required:

	Quantity	Units	Retail Cost per Unit	Total Cost	OEM multiplier	Total Cost
<b>Heat Exchangers</b>						
Exhaust pad	3.008512	ft3	\$ 12.00	\$ 36.10	0.3	\$ 10.83
Intake pad	2.918694	ft3	\$ 12.00	\$ 35.02	0.3	\$ 10.51
Liquid-to-liquid HX	165.5058	ft2	\$ 0.40	\$ 66.20	0.3	\$ 19.86
<b>Collector</b>						
plastic liner	74.2	ft2	\$ 0.40	\$ 29.69	0.3	\$ 8.91
cover	74.2	ft2	\$ 1.25	\$ 92.78	0.3	\$ 27.83
underlayment	74.2	ft2	\$ 0.15	\$ 11.13	0.3	\$ 3.34
frame	1.0	unit		\$ 100.00	0.3	\$ 30.00
<b>Storage</b>						
CaCl2	115.9	lbm	\$ 0.15	\$ 17.38	0.3	\$ 5.21
tank	27.8	gallons	\$ 1.00	\$ 27.76	0.3	\$ 8.33
<b>Pumps</b>	3	units	\$ 20.00	\$ 60.00	0.3	\$ 18.00
<b>Fans</b>	2	units	\$ 80.00	\$ 160.00	0.3	\$ 48.00
<b>Piping, controls, structure</b>				\$ 300.00	0.3	\$ 90.00
<b>Total Material</b>						<b>\$ 280.82</b>
<b>Factory Labor &amp; Overhead</b>						<b>\$ 200.00</b>
<b>Total Factory Cost</b>						<b>\$ 480.82</b>
<b>30% Mark Up</b>						<b>\$ 144.25</b>
<b>Freight</b>						<b>\$ 200.00</b>
<b>Installation</b>						<b>\$1,000.0</b>
<b>Grand Total</b>						<b>\$1,825.07</b>

Table 2 shows the projected cost for the solar air conditioner. For comparison, the installed cost of a conventional rooftop air conditioning unit is about \$2000 per ton, so it may be possible to achieve a first cost that is lower than that of existing systems.

Note that this analysis assumes 20% outside air is required. If the application requires 100% outside air, then the conventional air conditioner capacity would have to go up by about 67% because of the extra load. For these applications, the solar air conditioning cost per ton would be roughly 60% of the above values.

**Electrical Energy Use:** The electrical energy savings from the system are quite large. The primary energy use is related to the two fans and the main circulating pump. Assuming a 50% fan/motor efficiency the fan power is approximately 180 W for the example design shown above. The pump power should be significantly less and depends on the exact design for the system. The total power use is less likely to be less than .25 kW/ton. By comparison the typical energy use for a high efficiency (12 EER) rooftop unit is 1.0 kW/ton. The fan and pump power can be reduced further at the expense of increased size of the heat exchangers to provide a lower pressure drop. Also the use of variable-speed drives for the fans and pumps can greatly reduce the energy use.

For the California climate, little or no auxiliary power should be necessary to regenerate the desiccant. However, if unusual weather or load conditions demand it, desiccant material can be regenerated using off-peak (nighttime) electricity or by using a natural gas burner or other heater. The cheapest approach is to provide an electric resistance to provide backup to the solar collectors.

**Water Use:** Water use for the system is similar to that for evaporative coolers. For the example system the water evaporated is approximately 27.2 lbm/hr or 3.3 gallons per hour per ton of cooling. Additional water would be required to prevent salt buildup so approximately 4 gallons per ton per hour is reasonable estimate. For an annual operation 1500 equivalent full-load hours, the total water use is approximately 6000 gallons per year per design ton. At a cost of \$3/1000 gallons, the water bill is only \$18/ton/yr. By comparison, for an electric cost of \$.12/kwh, a conventional high-efficiency rooftop unit would use about \$180/yr/ton of electricity.

## **Conclusions**

1. Low-cost solar collectors can provide effective regeneration of desiccant liquid. Average evaporation rates of .5 to 1.0 lbm/ft<sup>2</sup>/day are easily achievable with simple open collector designs.
2. Several low-cost, high-performance liquid-to-liquid heat exchangers were tested with an overall heat transfer coefficients of between 20 and 40 Btu/hr/ft<sup>2</sup>/F. Leak-proof construction needs further work.
3. A 75% reduction in peak energy is feasible based on modeled results.
4. A thermal COP of 1.35 is achievable for a California climate.
5. Modeling shows that solar air-conditioning system has the potential to be competitive on a first-cost basis with conventional air conditioners.

## **Recommendations**

This work shows a great potential for solar air conditioners. Further work on the development of the cooler heat-transfer geometries is fundamentally important.

Heat exchanger design should:

1. Eliminate the possibility of circuit-to-circuit liquid-to-liquid leaks,
2. Use low-cost materials,
3. Have low maintenance and good reliability, and
4. Have compact, lightweight design.

The use of low-cost plastic material that eliminate corrosion problems with convention metal heat exchangers offer great promise for achieving the low-cost, high-performance designs that are essential for creating an economically viable solar air-conditioner.

## **Public Benefits to California**

As discussed earlier, the potential benefits energy saving benefits to California of solar air conditioning are enormous, amounting to billions of dollars in saved energy and generating

capacity. The results of this testing and analysis show that cost-competitive solar air conditioning is possible, although further work is necessary to achieve these benefits.

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US006513339B1

(12) **United States Patent**  
**Kopko**

(10) **Patent No.:** **US 6,513,339 B1**  
(45) **Date of Patent:** **Feb. 4, 2003**

(54) **SOLAR AIR CONDITIONER**

(75) Inventor: **William Leslie Kopko**, Springfield, VA (US)

(73) Assignee: **Work Smart Energy Enterprises, Inc.**, Washington, DC (US)

(\*) Notice: Subject to any disclaimer, the term of this patent is extended or adjusted under 35 U.S.C. 154(b) by 0 days.

(21) Appl. No.: **09/549,329**

(22) Filed: **Apr. 14, 2000**

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(51) Int. Cl.<sup>7</sup> ..... **F25D 17/06**

(52) U.S. Cl. .... **62/94; 62/271; 62/235.1**

(58) Field of Search ..... **62/271, 94, 235.1**

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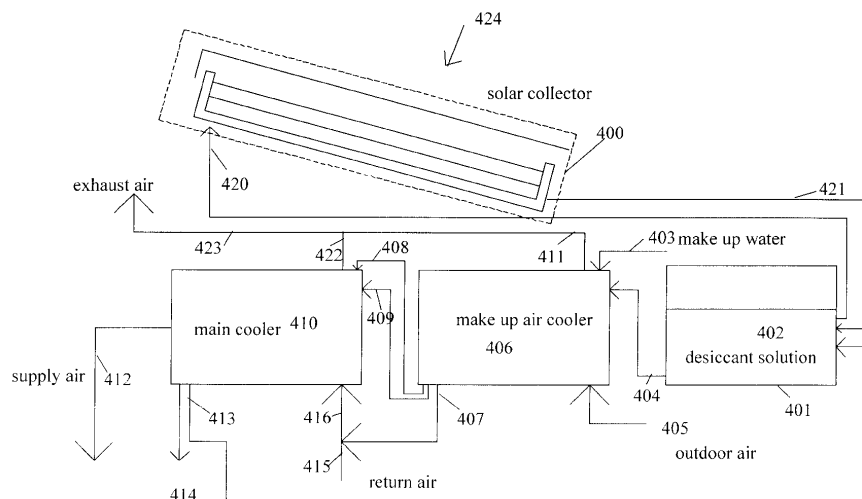
*Primary Examiner*—William C. Doerfler

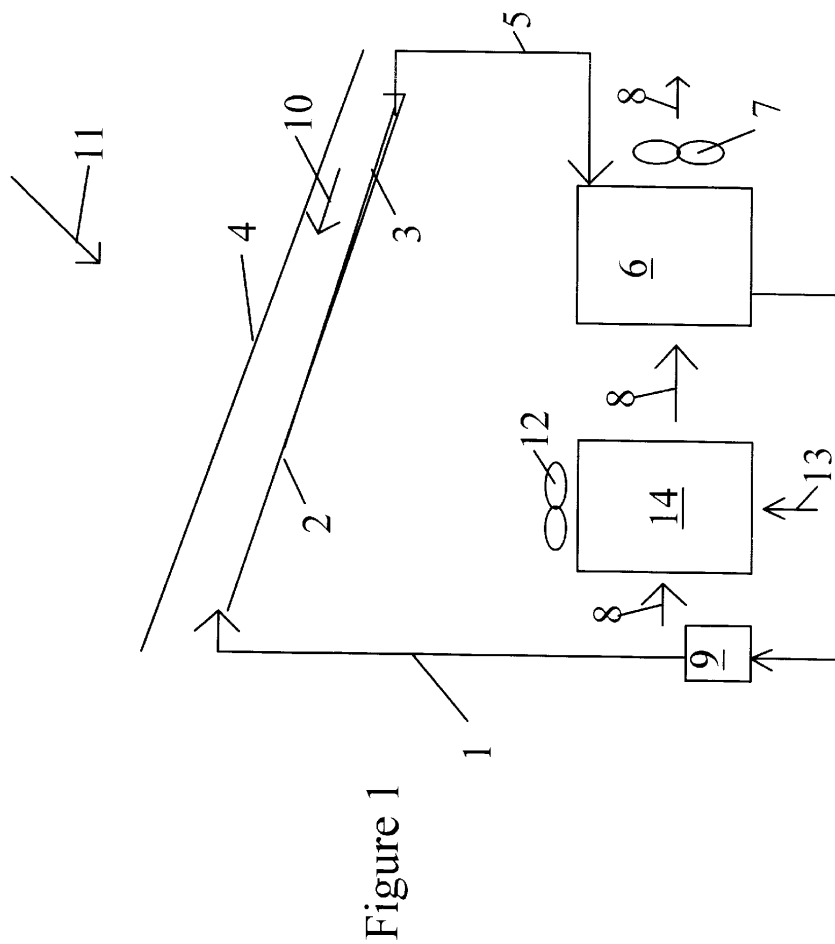
*Assistant Examiner*—Mark Shulman

(57) **ABSTRACT**

An air conditioning system that can be driven by solar energy. A solar collector used energy from the sun to evaporate water from a desiccant fluid. The desiccant fluid is then flows into a mass transfer device which removes moisture from an air stream. Calcium chloride is the preferred desiccant material and can serve as an energy-storage medium. Electric or fuel backup can be used with this system to regenerate the desiccant material. In some embodiments an indirect evaporative cooler is added to provide sensible cooling. A new desiccant cooling system that is specially designed to work with the properties of this desiccant and meet comfort requirements of conventional air conditioning.

**30 Claims, 15 Drawing Sheets**







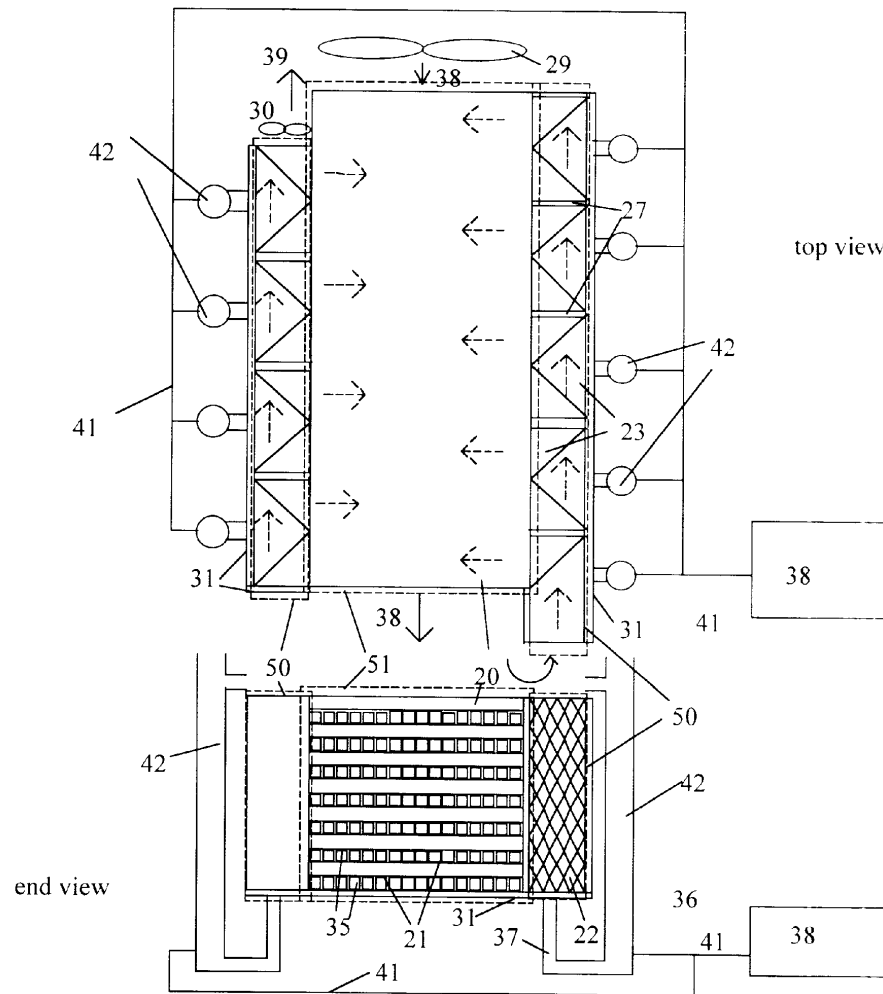
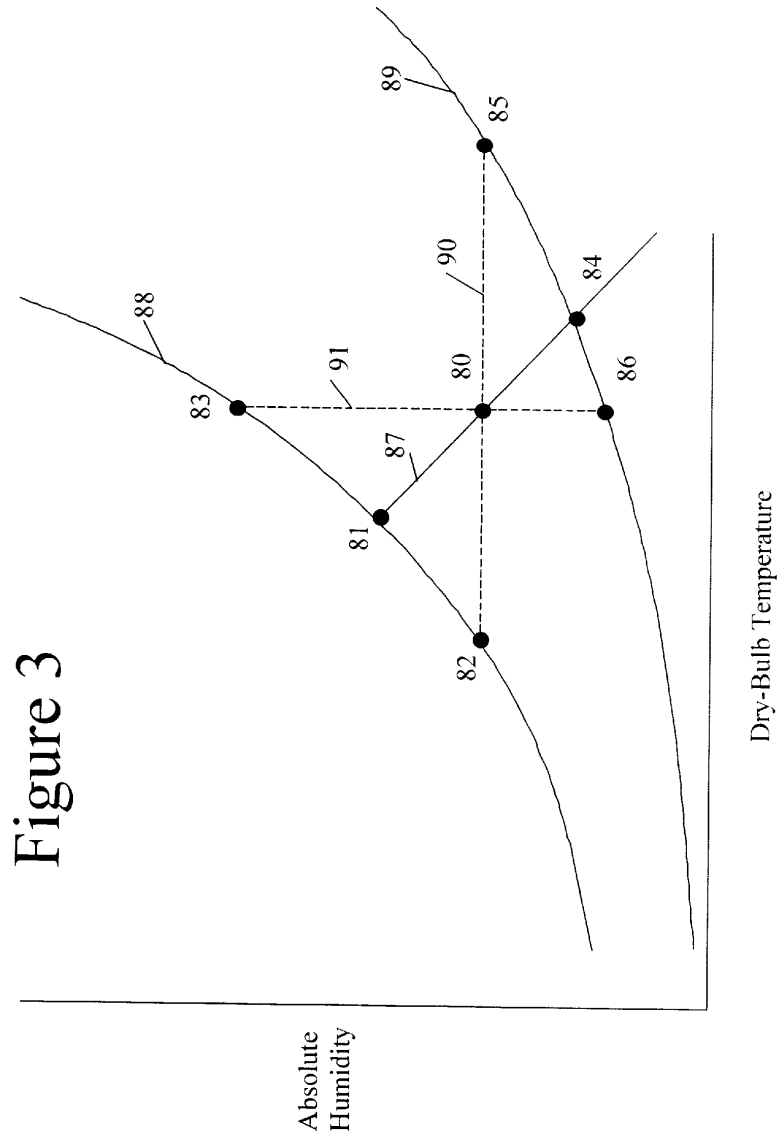


Figure 2



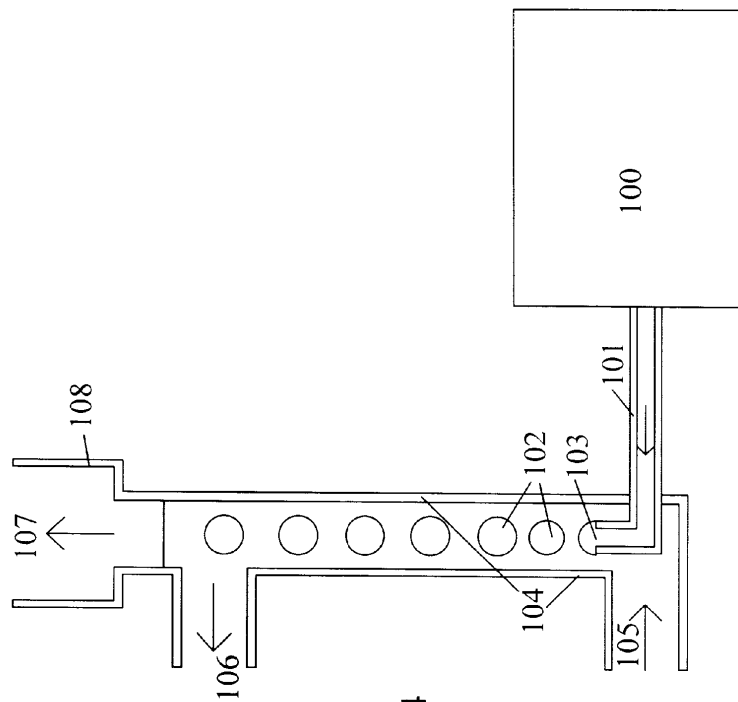


Figure 4

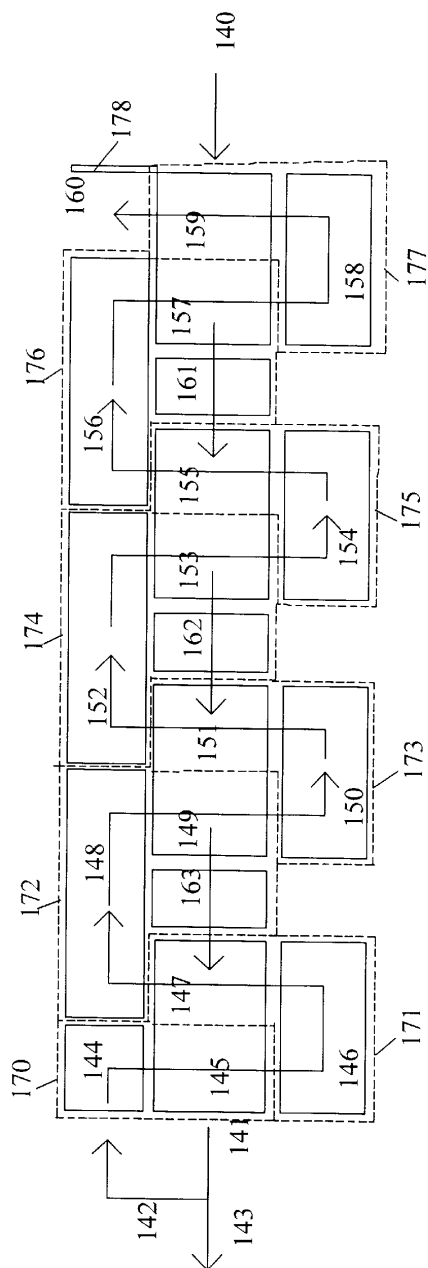
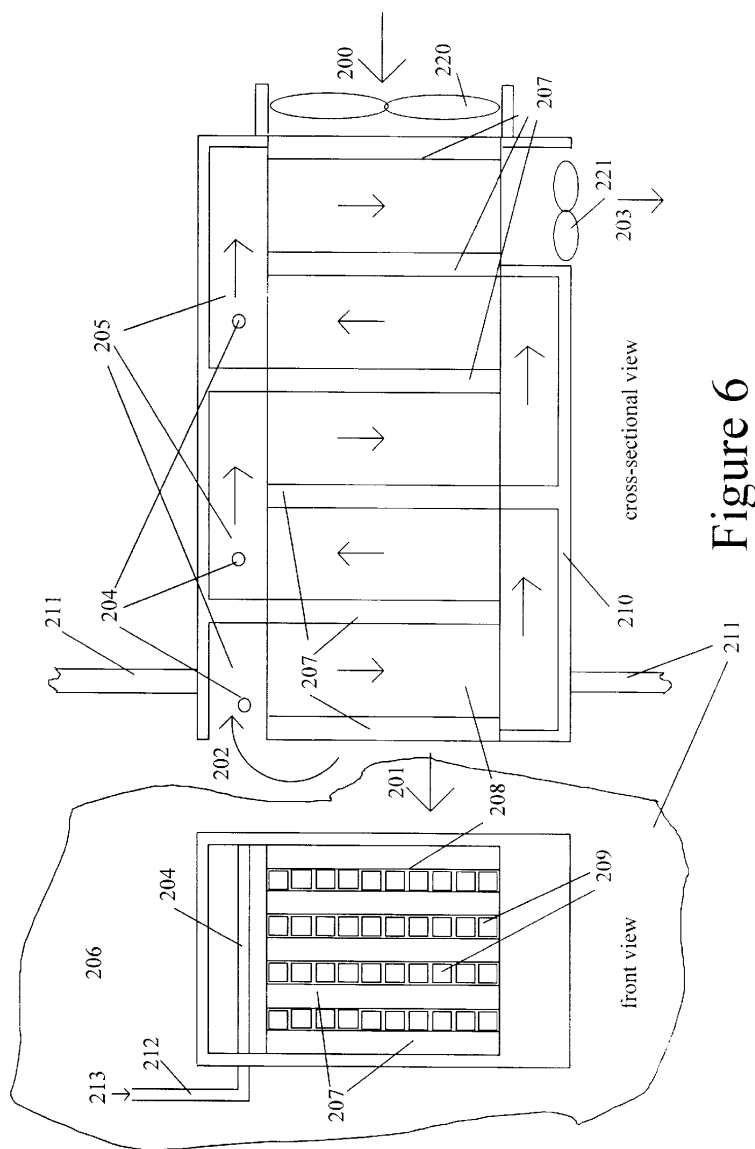


Figure 5



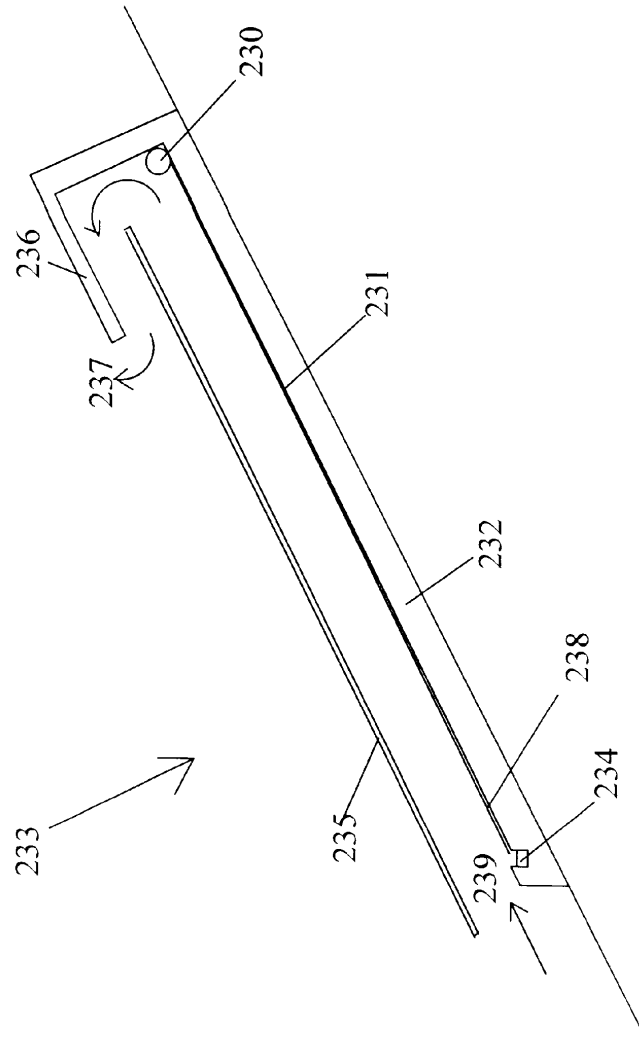
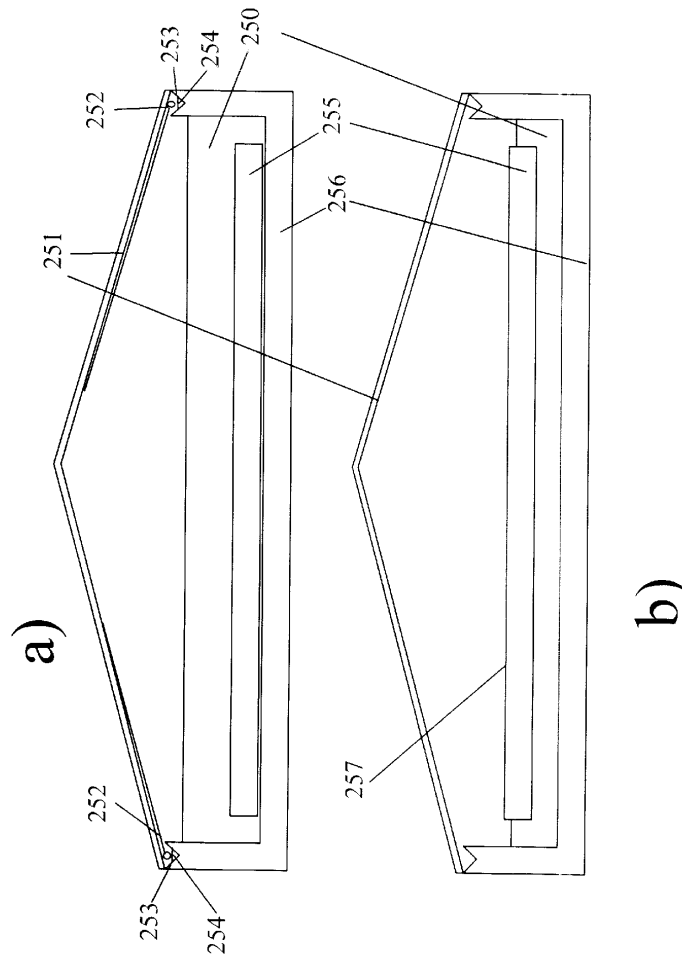


Figure 7

roof or supporting frame

Figure 8



## Figure 9

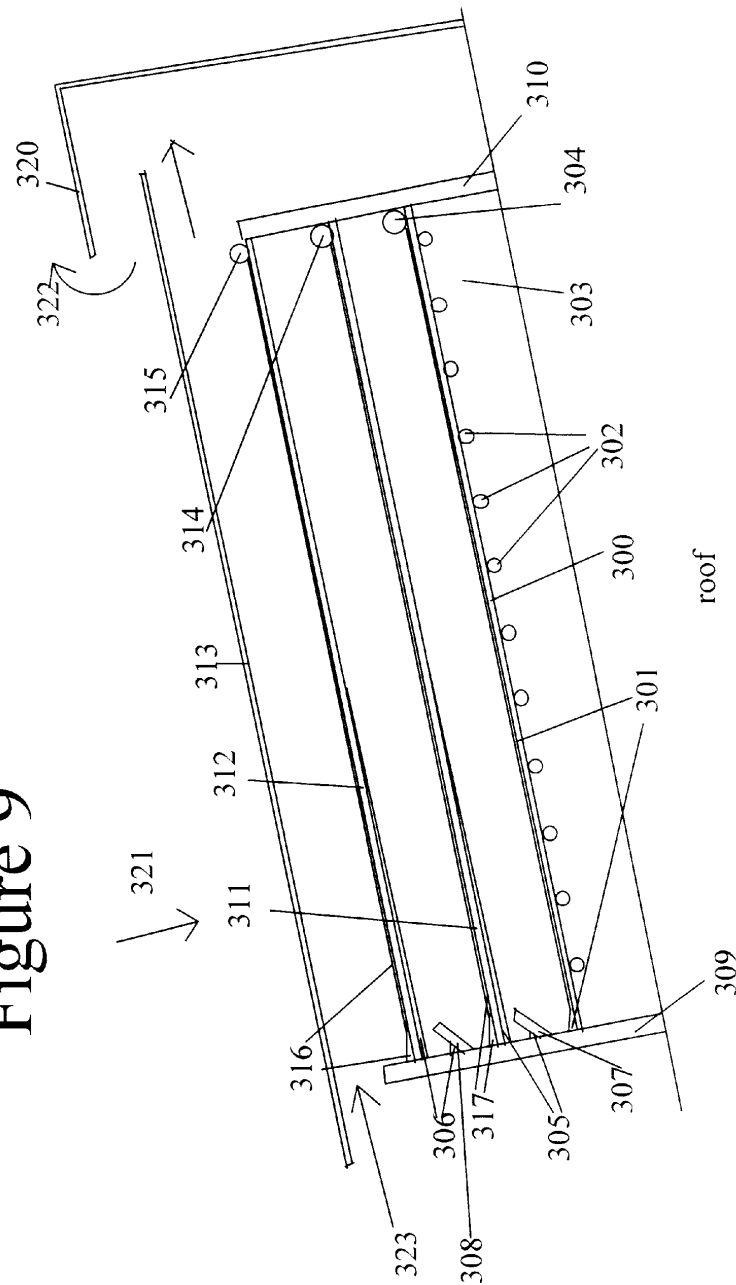
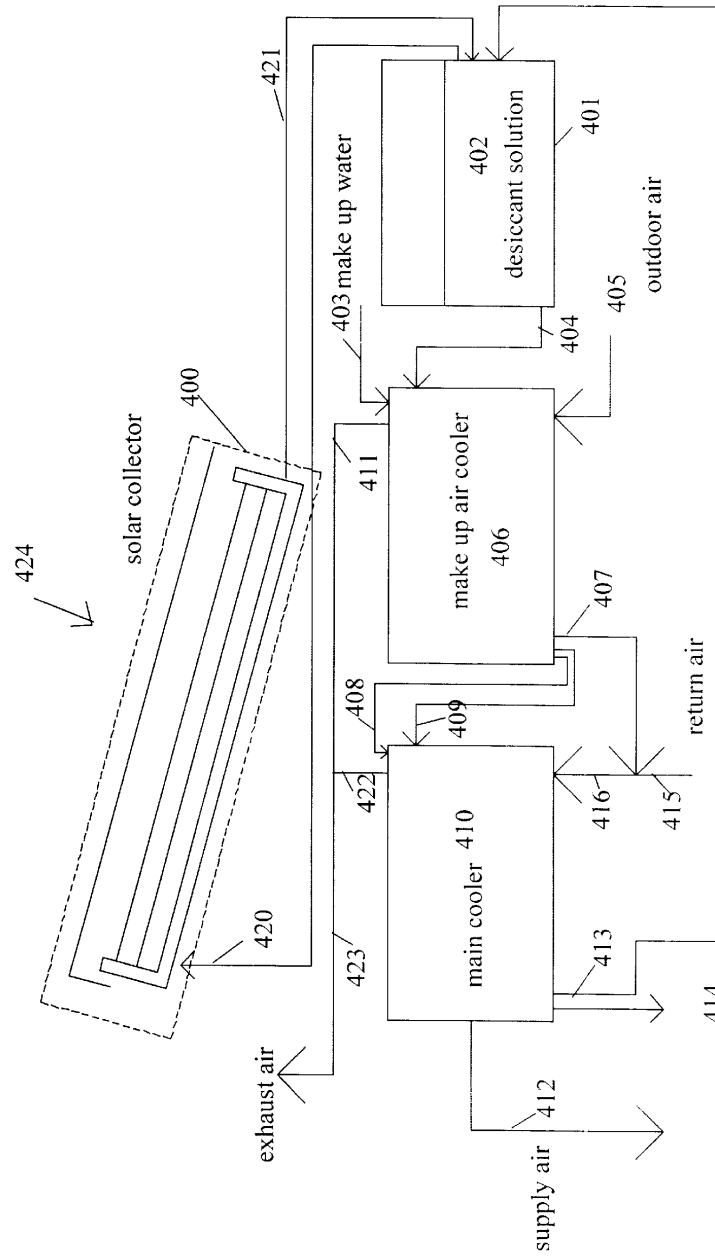




Figure 10



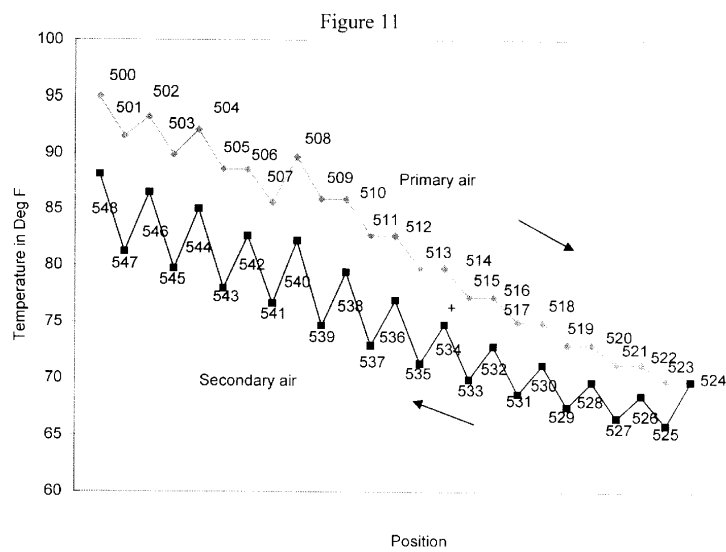
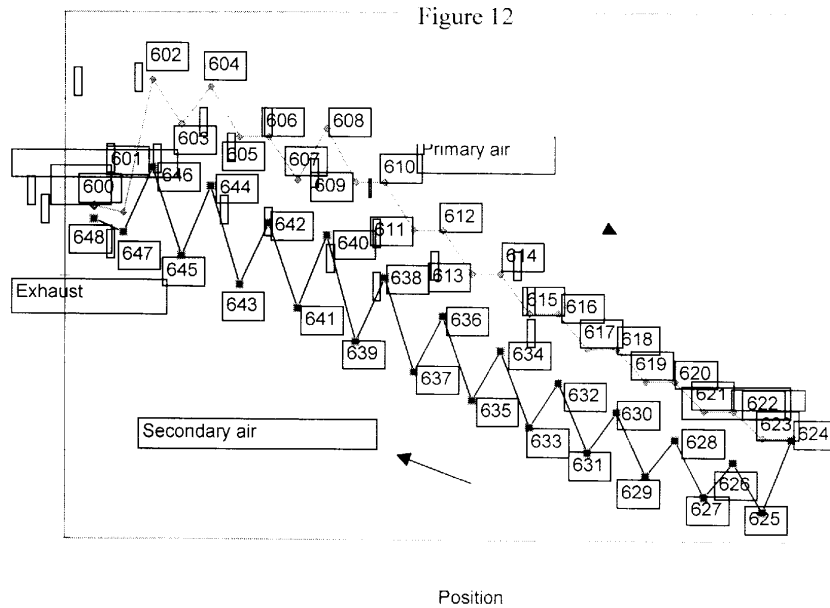


Figure 12



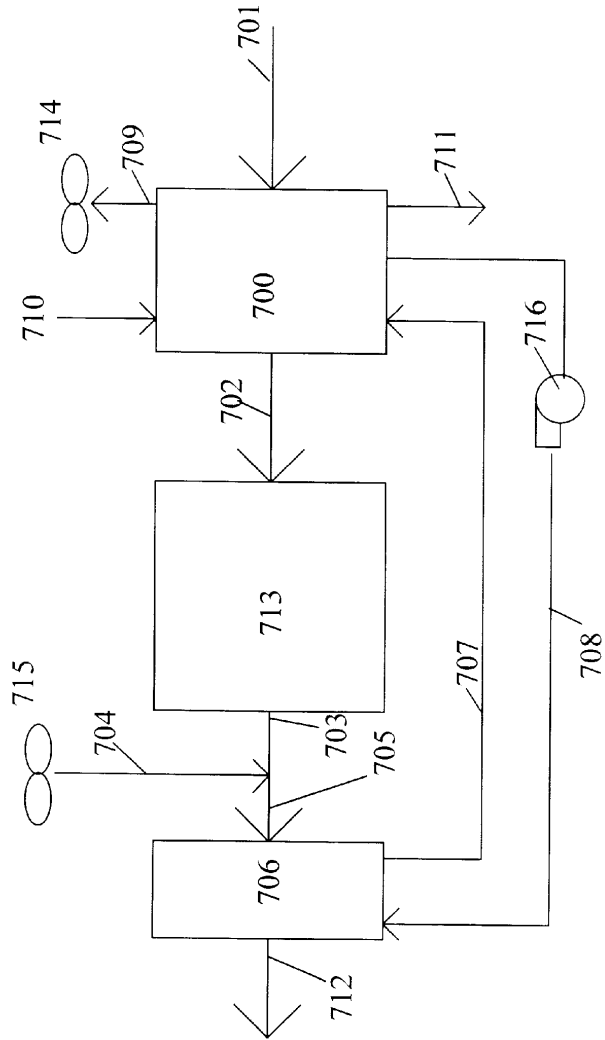


Figure 13

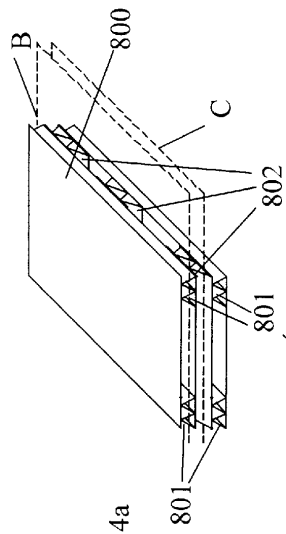


Figure 14a

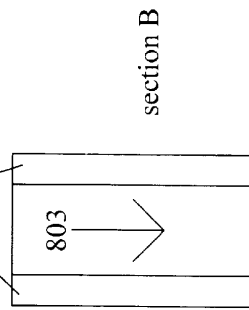


Figure 14b

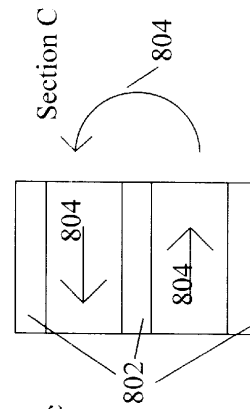
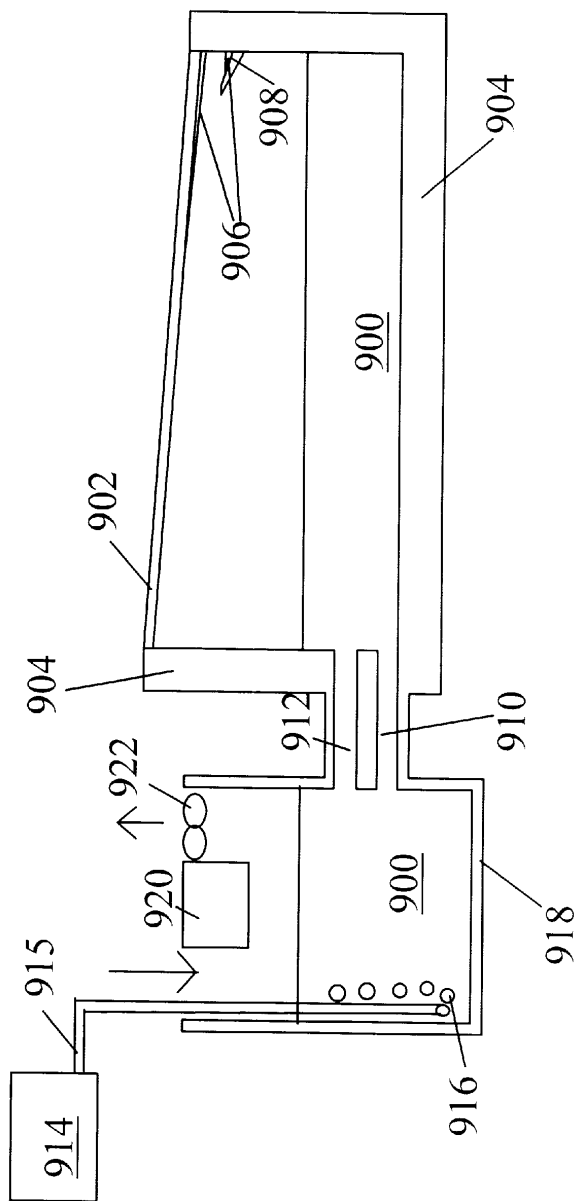


Figure 14c

Figure 15



US 6,513,339 B1

1

**SOLAR AIR CONDITIONER****CROSS-REFERENCE TO RELATED APPLICATIONS**

The applicant claims benefit of U.S. provisional application No. 60/129552 filed on Apr. 16, 1999.

**FIELD OF THE INVENTION**

This invention is in the field of air conditioning, specifically thermally driven air-conditioners that are capable of accepting a solar input.

**BACKGROUND OF THE INVENTION**

Solar air conditioning has great potential to reduce energy use from air conditioning. Sunlight is most plentiful in the summer when air conditioning is required.

The problem is that existing solar technologies have not produced systems that are economically competitive with conventional electrically driven systems. Prior work with solar air conditioning has not produced practical systems. Solar air conditioning systems have used two basic approaches in an attempt to capture the sun's energy for cooling—thermal and photovoltaic.

The photovoltaic systems use photovoltaic panels to convert solar radiation directly into DC electricity. Photovoltaic systems have two major advantageous attributes: they can use conventional electrically driven air-conditioning equipment which is widely available and inexpensive with the addition of the solar panels that use an inverter to produce AC power, and they can use the utility grid for backup power during dark or cloudy periods.

Unfortunately there are other attributes: the high cost of manufacturing, the low conversion efficiencies, and the need for a continual stream of photons to produce power, create three major disadvantages. First electricity from solar cells is very expensive because of the high cost of the solar panels. (Panels for a residential air conditioner can cost well over \$10,000.) Second the space needed for powering the air conditioning units is large. And third the panels provide no energy storage, which creates a need for use of grid based electricity at night and on cloudy day. In fact, the peak output from the solar panels occurs around solar noon, while peak air-conditioning load occurs several hours later, resulting in a significant mismatch between supply of needed power and demand. This mismatch greatly reduces the value of the system in reducing peak power demand to the utility, demands which recently deregulated markets is demonstrating are much more expensive to meet than had heretofore been obvious. For off-grid locations, the only viable energy storage system to match the provision of power to times when demand is high (later in afternoon and at night) is batteries. Batteries have a high first cost, require periodic replacement, and normally use toxic and/or corrosive materials. These problems have prevented the use of photovoltaic systems in other than a few high-cost demonstration systems.

Thermal systems use heat from the sun to drive an air conditioner. Typical approaches use a high-temperature flat-plate collector to supply heat to an absorption system. Systems with concentrating collectors and steam turbines have also been proposed. Natural gas or other fuel is used for backup heat. While thermal systems have the advantage of eliminating the need for expensive photovoltaic panels, they have attributes that produce major disadvantages.

One problem is the high cost and large size of the solar collectors. Flat-plate collectors running at about 190° F. (90°

2

C.) require double-glazing and selective surface to achieve reasonable efficiency levels, which greatly increases the collector cost. This high collector cost reduces the comparative attractiveness of such systems to standard vapor compression systems driven by grid electricity. Large collector size also reduces the potential market size by eliminating many locations from possible use of the systems.

Furthermore, existing thermal technologies also suffer from the poor COP of absorption systems, typically around 0.5. When combined with a typical collector efficiency of 20 to 50%, this inefficiency, besides creating a need for large collector areas, makes the whole system much less economically and environmentally attractive.

Another important problem introduced by the performance attributes of current solar thermal air-conditioning concepts is the high-cost and large size of high-temperature thermal storage. Large thermal storage is required to reduce backup energy (typically gas) that would be used much of the time when there was a mismatch between demand for cooling and solar inputs. This mismatch is the discrepancy between high solar input at noon and large demands for cooling during late afternoon, at night, and on cloudy days. A related problem with existing concepts for thermally driven solar cooling is the need for significant power input for circulating pumps and fans, which further reduces the possible energy savings.

Together these attributes for current concepts for thermally driven solar cooling imply that the large majority of their energy input would come from the backup fuel and electrical input for fans and pumps. In essence, these various problems mean that these solar systems are effectively very expensive gas-driven systems.

No commercially available or conceptually proposed system has been demonstrated that has the attributes that would be needed for commercially viable solar air conditioner. Commercial success will require the system to have the following attributes: low first cost (The market tends to be first cost driven so it is critical that the cost and thus ultimate selling price not be too high.); small collector area (critical to cost and to finding many locations in which installation is practical); small storage size (The mismatch between solar supply and cooling demand requires storage if the system is not to become a glorified means for using fossil energy and if it is to be practical to install in many locations—as well as low cost to manufacture.); easy to incorporate backup capability (Regardless of storage capacity, the ability of the system to meet demand in extreme and unusual circumstances will be critical to market acceptance; customers demand perfection and then some.).

Evaporative coolers are a related technology with a long history. Direct evaporative coolers are the simplest and most common. They consist of a means for moving air over a wet pad. Water evaporates from the pad and thereby cools and humidifies the air. They are commonly used for comfort cooling in warm, dry climates such as those found in the southwest U.S.

Indirect evaporative coolers are more sophisticated. An indirect evaporative cooler means that air is cooled by contact with a dry surface that is in turn cooled evaporatively.

Desiccant systems dry air for air-conditioning purposes. A typical system uses a solid desiccant impregnated on a wheel of corrugated metal or plastic.

Some more obscure systems appear in the patent literature, but each has its own problems. U.S. Pat. Nos. RE 20,469; 4,660,390; 4,854,129 describe regenerative indirect

## US 6,513,339 B1

3

evaporative coolers that use a portion of the air exiting the dry cooler as inlet air to the wet side. U.S. Pat. No. RE 20,469 describes a cumbersome arrangement of coils and cooling towers this complicated and expensive. U.S. Pat. No. 4,660,390 describes another system that uses tubes in a crossflow configuration to transfer heat between a wet side and a dry side. U.S. Pat. No. 4,854,129 also uses a system that uses a cooling coil with water from a cooling tower.

U.S. Pat. No. 5,050,391 describes another option for the desiccant system. This system uses solid desiccant material and a true counterflow arrangement for the heat exchangers. It also has essentially a single stage of cooling which limits its performance and its ability to use inexpensive desiccant materials.

## DESCRIPTION OF THE FIGURES

FIG. 1 shows a basic preferred embodiment of the invention.

FIG. 2 shows a design of an indirect evaporative cooler used in this invention.

FIG. 3 is a schematic psychrometric chart that shows how this cooler works.

FIG. 4 is drawing of a simple air-lift pump that is preferred for use in the invention.

FIG. 5 shows a diagram of a combination evaporative-desiccant cooler that is a component of this system.

FIG. 6 shows another cooler configuration that uses a water mist for evaporative cooling.

FIGS. 7, 8, and 9 shows solar collectors that may be used in the invention.

FIG. 10 shows a preferred embodiment of the invention.

FIGS. 11 and 12 are plots of temperatures through the coolers used in the invention.

FIG. 13 is an embodiment that can use exhaust heat from a gas turbine for cooling inlet air to the turbine.

FIGS. 14a, 14b, and 14c show details of heat exchanger design that may be used in the invention.

FIG. 15 shows another preferred embodiment that is suitable for use as a dehumidifier.

## DESCRIPTION OF THE INVENTION

## Description a Preferred Embodiment

FIG. 1 shows a preferred embodiment of the invention. A flow of desiccant fluid 1 is pumped by pump 9 to a solar collector 15 that acts as a regenerator for the desiccant fluid. The fluid trickles over a collector surface 2 in the form of a thin sheet 3. A cover 4 transmits solar radiation 11, which warms the desiccant fluid as it flows over the collector surface. A flow of air 10 removes water vapor that evaporates from the desiccant fluid. The concentrated desiccant 5 leaves the collector and flows to a mass-transfer device 6 that allows the desiccant to absorb moisture from an air stream 8. The mass-transfer device is preferably a direct-contact exchanger similar to those used for direct evaporative coolers and may also include a pump for recirculating the desiccant liquid through the device to ensure good mass transfer. A supply air fan 8 moves the air stream through the mass-transfer device.

An indirect evaporative cooler 14, cools the air stream 8 without adding moisture to it. A fan 12 draws a secondary air stream 13 through the cooler. The secondary air stream may be exhaust air from a building, ambient air, or a portion of the conditioned air leaving the evaporative cooler or mass-

4

transfer device. This indirect evaporative cooler is optional and may be eliminated in cases where no sensible cooling is required.

## Indirect Evaporative Cooler Design

FIG. 2 shows the one heat exchanger system that is suitable for use as an indirect evaporative cooler for this system. The cooler has two basic parts—mass transfer means 50 and an air-to-air heat exchanger 51. The cooler is shown without a top cover for clarity. Corrugated panels 20 for secondary air are oriented so that corrugations run from side to side while corrugated panels 21 for primary air have corrugations that run from end to end. Channels 35 formed by the corrugations in panels 21 allow for free flow of air through the panels. Likewise similar channels run in a perpendicular direction through panels 20. The panels 20 are stacked alternately with panels 21 so that the channels for each panel are perpendicular to the channels for the adjacent panels. The outside surface of the sheets may be covered with an adhesive or filler material to ensure good contact between the sheets. For maximum durability the panels are preferably made of polypropylene or polyethylene plastic. Metals, such as aluminum, are also possible materials.

Another option is to use corrugated cardboard and paper. Waterproof adhesive material, such as one based on acrylic or linseed oil, can coat the paper or cardboard and joins the layers together to form a single unit. The advantage of cardboard or paper is its very low cost. The disadvantage is that it may be less durable. One advantage of this system is that it is not possible to create condensation within the heat exchanger, which allows the possible use of cardboard in some applications. This is especially true in desiccant applications since it is possible to keep both air streams above their respective dew point temperatures even when outdoor conditions are at 100% relative humidity (such as rain or fog conditions).

The main fan 29 moves primary air stream 38 through a single pass through channels 35. The stacked panels 20 and 21 form a heat exchanger that cools the primary air without addition of humidity. A portion of the cooled primary air splits off and forms a secondary air stream 39 which is moved by secondary fan 30. The secondary air first flows through multiple passes of the air-to-air heat exchanger to cool the primary air stream. The direction of the secondary air flow through the heat exchanger is shown by the dashed arrows.

The passes of the secondary air stream are preferably arranged in a counter crossbow configuration with a mass transfer means ahead of each pass of the heat exchanger. The mass transfer means is preferably a direct evaporative cooler. The direct evaporative cooling sections 23 form U-bends that direct the secondary air through each pass. As shown in the figure three triangular pieces fit together to form two mitered elbows which make each U-bend. Pass dividers 27 would normally be included to prevent excessive leakage between passes in the wet media in each pass. Housing 31 ensures that excessive air does not leak in or out of the heat exchanger.

The chief use of this system is as an evaporative air cooler, but many other applications are possible. In addition to air, this system can work equally well with any number of nonreactive gasses such as nitrogen, carbon dioxide, inert gasses, etc. This system can also be used as a heater. For example if a desiccant liquid is substituted for water and the entering gas stream has a high relative humidity, the system would act to heat the gas stream. Volatile liquids other than



5

water-based solutions can be used in the system, but they are very expensive and may pose risks with flammability or toxicity.

The direct evaporative cooling sections need to be thoroughly wetted to ensure good evaporation while minimizing mineral deposits. In addition there is normally a large change in the wet-bulb temperature from one end of the heat exchanger to the other, so that water circulation between passes needs to be minimized to reduce undesirable heat exchange. These factors make it desirable to use multiple water circuits with multiple pumps.

For large systems using multiple pumps does not introduce a significant cost penalty, but for small systems multiple pumps can add greatly to the cost. One possible solution is to have multiple pumps that share a common shaft and motor. Seals separate the pumps from each other to minimize leakage and heat transfer.

FIG. 2 shows another possible option for circulating liquid using air-lift pumps 42. Air pump 38 supplies pressurized air through air line 36. Drain 37 removes water from the bottom of the direct evaporative cooling sections 23. Air bubbles into the water to create a pumping action. Extra water can be supplied to the pumps to make up for that lost to evaporation or blow down.

Other configurations of the air-to-air heat exchanger are possible. For example instead of stacking corrugated panels on top of each other, it may be possible to use spacers between the panels that are oriented in the same direction. The spacer could separate the passes of the secondary air and allow free flow of the secondary air over the panels. In this configuration the primary air would flow inside the channels of the panels. This alternative configuration should reduce material cost and reduces thermal resistance of the walls between the two air streams.

Another configuration would simply stack sheets with spacers to direct air flow. For example sheets of paper can be separated by corrugated cardboard spacers. The spacers would be on the order of 0.1 inches thick to form a flow channel for air. The orientation of the spacers would alternate so that the air flow for the secondary air is perpendicular to the that for the primary air. This arrangement would use a minimum amount of material and is a simple design and would be the preferred configuration for materials, such as paper, that are easily glued together.

#### Indirect Cooler Theory of Operation

FIG. 3 is a psychrometric diagram showing how the idealized behavior of the system. For the case of conventional direct evaporative cooler, the process start at entering air 80 and follow the constant wet-bulb temperature line 87 (which is also essentially a line of constant enthalpy) and approach ideal exit condition 81 which is along saturation curve 88. For the new system used as a cooler there are two exit conditions, the supply air 82 and exhaust air 83. The primary air stream follows the line of constant absolute humidity 90 and approaches the saturation condition at point 82. A portion of this air exits system as supply air and the rest moves along the saturation line 88 as it is heated and humidified until it approaches the ideal exhaust condition 83. This exhaust condition is ideally at the intersection of the saturation line 88 and the constant dry-bulb temperature line 91. The result is a colder supply air temperature than is possible with a simple direct evaporative cooler.

For the case of a heater, the conventional direct contact system would again follow constant wet-bulb line 87. The process would start at the entering air condition 80 and

6

approach ideal equilibrium point 84, which is on the desiccant equilibrium curve 89. For the new system, there are again two ideal exit conditions, the supply air condition 85 and the exhaust condition 86. As with the cooler, the ideal supply air temperature is the point along the constant absolute humidity line 90 that is in equilibrium with the liquid. In both cases only a fraction of the primary air stream needs to be exhausted, typically 30 to 50%, which leaves the rest as supply air.

#### Air-Lift Pump

FIG. 4 shows detailed drawing of an air-lift pump that is suitable for pumping liquid desiccant and water. Air pump 100 supplies pressurized air 103 to air line 101. The air pump is preferably an aquarium pump or similar design. The air line 101 discharges inside water pipe 104 and creates a flow of air bubbles 102. The bubbles lower the average density of the fluid column which causes the air and water mixture to move upward. This upward movement draws intake water 105. On the top end a separator 108 allows outlet air 107 and outlet water 106 to discharge from the pump in separate flows. The advantages of this pump include low cost, simple design, reliability, no moving parts. This pump is excellent for handling small liquid flows with a small head requirement. A single air pump can drive many air-lift pumps and thus create many liquid circuits.

#### Evaporative-Desiccant Heat Exchanger

FIG. 5 shows a heat exchanger that adds the use of a desiccant. This arrangement has eight stages of cooling, 170, 171, 172, 173, 174, 175, 176, and 177. The incoming primary air 140 enters the rightmost stage 177. It then flows through the eight stages in a straight path where it is cooled and dehumidified. The exiting primary air 141 splits into two flows. A secondary flow 142 goes back through the heat exchanger in a counter-crossflow arrangement. The remaining primary air 143 is supplied to the load.

As shown in this figure each stage of cooling includes an evaporative pad for cooling and humidifying the secondary air and an air-to-air heat exchanger that transfers heat between the two air streams. In addition some of the stages include desiccant, which dries the primary air stream. The desiccant is preferably a liquid, such as an aqueous solution of calcium chloride, lithium chloride, lithium bromide, glycol, or similar material. Materials such as sodium hydroxide and sulfuric acid have excellent physical properties but are very corrosive and dangerous to handle. Calcium chloride is very inexpensive, has acceptable thermodynamic properties, has relatively low toxicity, and is generally the preferred desiccant material for this system.

Starting with the rightmost stage 177 secondary air flows over evaporative pad 158 and through air-to-air heat exchanger 159. The primary air flows through the other side of the air-to-air heat exchanger 159. The flow directions of the two air streams are perpendicular to each other with the primary air going is a straight line through the heat exchanger.

Next to the left is stage 176 which includes a desiccant 161 that is in the primary air stream. The desiccant is preferably provides a surface that is wetted by a liquid desiccant that is in direct contact with the primary air stream. The evaporative pad 56 cools and humidifies the secondary air stream. The secondary air stream cools the primary air stream in air-to-air heat exchanger 157.

The stages 170, 175, 173, and 171 are similar to stage 177 with evaporative pads 154, 150, 146, and 144 in the sec-

ondary air stream and air-to-air heat exchangers **155**, **151**, **147**, and **145** transferring heat between the two air streams.

The stages **174** and **172** are similar to stage **176**. They include desiccants **162** and **163** which dehumidify and increase the temperature of the primary air stream. The evaporative pads **148** and **152** cool and humidify the secondary air which cools the primary air in air-to-air heat exchangers **149** and **153**.

#### Mist Cooling Option

FIG. **6** is another heat exchanger configuration that uses a mist cooling system. Fan **220** draws in the main air stream **200**. The air moves in a straight line through interior channels **209** in panels **208** and is cooled by evaporating water mist **205** on the outside of panels **208**. The mist is supplied by nozzles **204** that are connected by way of pipe **212** to a source of pressurized water **213**. The water is preferably demineralized and filtered to prevent clogging and fouling of the nozzles and the heat exchanger surfaces. As the main air stream leaves the heat exchanger, a portion of the air forms a secondary air stream **202** which returns on the wet side of the heat exchanger. Dividers **207** direct the secondary air in multipass counter crossflow arrangement as shown by the arrows. Housing **210** and wall **211** prevent undesirable air leakage. Fan **221** moves the secondary air out of the heat exchanger in exhaust stream **203**. Drains **222** may be included at the bottom of the housing to remove excess water.

#### Simple Solar Collector

FIG. **7** is a simple solar collector for regenerating a desiccant solution. Liquid header, **230** trickles desiccant liquid **238** over collector surface **231** which is tilted at an angle to allow drainage through trough **234**. Solar radiation **233** is transmitted through cover **235** and warms the desiccant liquid **238**. The collector surface is preferably black and is backed by thermal insulation **232** to maximize energy collection.

The cover is preferably of a transparent material such as polycarbonate, polyvinyl chloride (pvc), fluoropolymers (such as Tedlar), acrylic, or other plastic. For rigid materials, the cover may be flat or corrugated. Flexible films such as Tedlar would normally be held taught in a frame. Glass is another option for a cover material. The selection of optimum collector material depends on the cost and durability of the different materials. The duty is similar to that for greenhouses, windows, skylights, etc. with temperatures that are much lower than those for most other types of solar collectors.

Incoming air **239** flows by natural convection between the cover and the collector surface and absorbs moisture that evaporates from the desiccant liquid. The leaving air **237** exits the collector between end piece **236** and cover **235**. The end piece is shaped so as to prevent rain from entering the collector.

The desiccant solution **238** preferably flows in a sheet that wets the entire collector surface. One way of achieving this flow is to use a screen, cloth, or other roughness on collector surface **231**. Another option is to use a relatively large flow of liquid to create a continuous film of liquid. A third option is to add a detergent or other wetting agent to the solution to enhance wetting. Combinations of these three alternatives are also possible.

The orientation of the collector is preferably such that the rays of midday summer sun is approximately normal to the

collector surface. For the most of the US this corresponds to a tilt of angle of 5 to 30 degrees from horizontal towards the south. In tropical areas the collector surface can be nearly horizontal with only a few degrees of tilt to allow adequate drainage. In the Southern Hemisphere the collector is preferably tilted to face north.

The operation of this collector is quite simple. When solar radiation is available to raise the collector surface to a temperature that is sufficiently high, desiccant liquid is allowed to flow through the collector. At other times no liquid would flow. A simple thermostat that controls the circulating pump can accomplish this control.

This collector has several advantages. First the temperatures necessary to regenerate the desiccant liquid are quite low, in the range of about 110 to 140 degrees Fahrenheit, which allows the use of inexpensive materials such as plastic, wood, asphalt roofing material, etc. Second the operation is very simple with no moving parts. Third the collector can be mounted on an existing roof or other surface.

While the preferred embodiment of the solar collector includes a cover, the collector would also function without a cover. The main advantages eliminating the cover are reduced cost and complexity. The collector can, in fact, be as simple as a section of dark roof or other surface with the addition of a system desiccant liquid over the surface. The chief problem with operation without a cover is that rain would tend to wash away any residual desiccant solution. The resulting diluted desiccant would have to be discarded or else it would dilute the solution in storage. Wind or leaves may also carry desiccant solution away when no cover is present. Loss of large quantities of desiccant solution is costly, may damage nearby plants or metals, and may create unsightly salt deposits on surrounding surfaces. A simple cover should greatly reduce or eliminate these problems, but it is not absolutely necessary for operation.

In dry climates an evaporation pond is an alternative to a solar collector. A pond is an inexpensive way of regenerating a desiccant solution. The chief problems are related to control over the salt concentration. An extended rainy period can dilute the solution excessively, while long periods of dry, sunny conditions can result in crystallization. Another issue is the possibility of high winds blowing droplets of desiccant solution onto surrounding surfaces which may create problems with corrosion, plant damage, etc.

#### Solar Still with Automatic Shut-Off Feature

FIGS. **8a** and **8b** shows a solar still with an automatic shutdown feature that can regenerate the desiccant liquid. FIG. **8a** shows the still in normal operation. Solar radiation **260** warms desiccant liquid **250**. Water vapor evaporates from the desiccant liquid and form condensate **252** on cover **251**. The condensate trickles down the inside of the cover and collects in troughs **253** and **254**. An insulated tank **256** forms the bottom and the sides of the collector and holds the desiccant liquid.

Float **255** provides a simple control mechanism. As shown in **7a**, the desiccant solution is relatively dilute, which reduces its density and causes the float to sink to the bottom of the tank **256**. FIG. **8b** shows a situation where the desiccant solution is quite concentrated, which increases its density and causes the float to rise to the top of the pool of desiccant liquid. Water evaporates out of any remaining desiccant liquid on the surface of the float and eventually creates a thin layer of salt crystals **257**, which helps to reflect solar energy and controls the temperature inside the collec-

tor. The action of the float thus provides an automatic shutdown feature that prevents excessive crystallization of the desiccant solution.

The float should be of nearly neutral buoyancy with respect to the desiccant solution, so that the change in solution density is enough to determine whether the float rises or sinks. The float materials should be resistant to high temperatures and compatible with the desiccant solution. Foam glass, ceramics, high-temperature plastics, and metals that are compatible with the desiccant are likely choices. The float may be divided into smaller pieces to simplify handling.

The sealed cover has the advantage of keeping ambient moisture out of the desiccant during dark or cloudy periods. A cover without a seal would allow free movement of humid air, which can add undesirable moisture to the desiccant solution.

#### High-Performance Solar Collector with Electric Back-Up

FIG. 9 is a high-performance solar collector with electric backup for use in regenerating the desiccant liquid. This collector provides three stages of regeneration and would operate with peak temperatures of around 200 to 240° F. The three stages of regeneration are arranged so the waste heat from higher-temperature stage drives a lower temperature stage. Desiccant liquid 301 flows from bottom header 304 over collector surface 300. Electric heater elements 302 are located just under the collector surface and provide an auxiliary source of heat. Insulation 303 prevents excessive heat loss through the back of the collector.

The collector has three covers. The bottom cover 311 and middle cover 312 fit tightly with frames 309 and 310 to minimize air leakage. The top cover 313 has large gaps at each end, which allow for air movement under the cover. The bottom and middle covers 311 and 312 are preferably of glass or other heat resistant material. The top cover is preferably made of a tough plastic material to minimize risks of hail damage or other hazards. The top cover experiences much lower temperatures, so heat resistance is not an important issue.

Covers 311, 312, and 313 transmit solar radiation 321 which warms collector surface 300. At night or during cloudy periods, electric heater elements 302 provide an auxiliary heat source for warming the collector surface. The warm temperatures cause moisture in the bottom stream of desiccant liquid 301 to evaporate. The water vapor thus produced is moved by convection and/or diffusion to bottom cover 311 where it condenses to form condensate 305. The condensate flows down the undersurface of bottom cover and collects in a trough formed by catch member 307 and frame 309. The condensate then flows out of the collector and can be used in an evaporative heat exchanger, as distilled drinking water, or discarded. Likewise a portion of the desiccant liquid that collects at the bottom of the collector can be returned to storage and the rest can be recirculated.

The middle stream of desiccant liquid 317 flows from middle header 314 over the top of the bottom cover 311. The heat transmitted to the bottom cover from below evaporates moisture from the middle stream of desiccant liquid 317. The moisture condenses on the bottom surface of the middle cover 312 to form the middle condensate stream 306 which drains through the trough formed by catch piece 308.

The top header 315 supplies the top desiccant liquid stream 316 that flows across the top surface of the middle

cover 312. Moisture that evaporates from the top desiccant liquid stream 316 is removed by natural convection of air and does not normally condense on the top cover 313. Entering air 323 flows through the collector and receives the evaporating water vapor. End piece 320 prevents excessive amounts of rain from entering the collector and air leaves the collector as exhaust stream 322.

While this figure shows an electric heater as the backup heating system other heat sources are possible. Hot water, steam, and direct heating with a fuel are also possible. The surface may be heated directly or the desiccant liquid can be heated in a separate heat exchanger. For small systems, a gas water heater may provide the heat source. The selection of the heat source would be determined by fuel cost and availability, installed cost and other factors. If electric power is used it would preferably be used at night to take advantage of lower off-peak electric rates.

The optimum collector temperatures depend on the desiccant concentration, the ambient conditions, and other factors. For calcium chloride a minimum temperature difference of about 30° F. is necessary to evaporate a desiccant solution and condense the resulting water vapor. Assuming a temperature difference of about 40° F. in each stage, temperature of the middle cover would be about 120° F., the bottom cover would be 160° F. and the collector surface would be 200° F. If the peak temperature is a problem, a two-stage system can be used instead but with a performance penalty. Of course four or more stages are also possible, but the collector temperature is normally limited to the boiling point of the desiccant liquid which would be roughly 230° F. or somewhat higher.

This collector requires a means for circulating the desiccant liquid. Air-lift pumps or conventional pumps are possible alternatives. The desiccant liquid would normally be recirculated several times through the collector before returning to the storage tank. This recirculation ensures sufficient movement of liquid for adequately wetting the surfaces inside the collector without creating excessive heat loss. A heat exchanger between liquid entering and leaving the collector would further reduce heat losses, but this feature is not required for operation of the system.

The covers need to be tilted by roughly 10 degrees or more to ensure proper drainage of condensate. Smaller angles could result in excessive drainage of condensate back into the desiccant, which would create a large performance penalty.

The actual collector surface can be horizontal, which can allow desiccant to pool inside the collector. This arrangement allows the collector to also function as storage tank. Using a float as a shut-off control as shown earlier would be desirable for this system. This configuration may be especially desirable in tropical areas where the sun's rays are nearly vertical during much of the day.

#### Preferred Embodiment

FIG. 10 is a schematic drawing of a complete solar air-conditioning system. Solar collector 400 concentrates desiccant solution using heat input from solar radiation 424 or auxiliary heat source as explained in the description of FIGS. 8A and 8B. A tank 401 stores a large quantity of desiccant solution 402. The weak desiccant solution 420 leaves tank 401 and flows through the solar collector 400 and returns as a concentrated desiccant solution 421.

The amount of desiccant solution in the tank depends on the size and efficiency of the system and the length of operation required. A reasonable objective would be to

achieve one to three days of storage capability. For a system with a rated capacity of 12,000 Btu/hr system with a 50% duty cycle over two days, corresponds to storage requirement of 288,000 Btu of cooling (24 ton-hours). For a cooling COP of 1 and heat of vaporization of 1000 Btu/lbm mean that this cooling requirement could be met by the ability to absorb 288 lbm of water vapor. For calcium chloride solution with a starting concentration of 50% and an ending concentration of 40%  $\text{CaCl}_2$  by weight, requires two pounds of calcium chloride to absorb one pound of water. This analysis means that 576 lbm of calcium chloride is required to store the required cooling. For a 40% final concentration, this corresponds to a tank capacity to handle 1440 pounds of solution or about 150 gallons.

The energy storage density per unit volume is almost twice that of ice and requires no special insulation. With proper sealing, the tank can store this cooling capacity indefinitely with essentially zero loss. The cost of calcium chloride is on the order of \$.20/lbm so that the cost of the salt for the above example is a little over \$100. The cost of storage tank is similar. These costs work out to be roughly \$10 per ton-hour, which is roughly 10 to 20% of the cost of conventional ice storage or cold-water storage. This storage system thus has major cost and performance advantages compared to other systems. This inexpensive, compact storage capability combined with simple, efficient solar recharging is a tremendous improvement over the prior art.

#### Theory of Operation of the Coolers

FIG. 11 shows how temperatures vary through a system for supplying outside air. The layout of the modeled system is similar to that shown in FIG. 4. This system has 12 stages. Each stage has an air-to-air heat exchanger between primary and secondary, a direct evaporative cooling section on the secondary side, and a desiccant section on the primary side. In this analysis only stages 1, 2, and 4 have active desiccant sections. The mass flow rate on the secondary air stream is approximately half of that of the primary air stream for this analysis. The entering air conditions are 95° F. dry bulb and 75° F. wet-bulb temperature, which is a typical design condition for the eastern US.

The system in FIG. 11 can be used to supply outside air to laboratories or other applications that have a large outside air requirement and limitations on heat recovery or other energy-saving technologies. It has the advantage of conditioning outside air with extremely high efficiency and requires no access to exhaust air. Changes in the details of the design can give different supply-air conditions as required for a particular application.

TABLE 1

Primary Air Stream:				
Location	Temperature (degrees F)	Enthalpy (Btu/lbm)	Stage	Location
500	95.0	38.6	1	Inlet
501	91.6	37.8	1	After heat exchanger
502	93.3	37.8	1	After desiccant
503	89.9	37.0	2	After heat exchanger
504	92.1	37.0	2	After desiccant
505	88.6	36.1	3	After heat exchanger
506	88.6	36.1	3	After desiccant
507	85.6	35.4	4	After heat exchanger
508	89.7	35.4	4	After desiccant
509	85.9	34.5	5	After heat exchanger
510	85.9	34.5	5	After desiccant

TABLE 1-continued

511	82.6	33.7	6	After heat exchanger
512	82.6	33.7	6	After desiccant
513	79.8	33.0	7	After heat exchanger
514	79.8	33.0	7	After desiccant
515	77.2	32.3	8	After heat exchanger
516	77.2	32.3	8	After desiccant
517	75.0	31.8	9	After heat exchanger
518	75.0	31.8	10	After desiccant
519	73.0	31.3	10	After heat exchanger
520	73.0	31.3	10	After desiccant
521	71.3	30.8	11	After heat exchanger
522	71.3	30.8	11	After desiccant
523	69.9	30.5	12	After heat exchanger
524	69.9	30.5	12	After desiccant (supply air)

Secondary Air Stream:

Location	Temperature (degrees F)	Enthalpy (Btu/lbm)	Stage	Location
525	88.1	45.9	12	After direct evaporative cooler
526	81.2	44.3	12	After heat exchanger
527	86.5	44.3	11	After direct evaporative cooler
528	79.7	42.7	11	After heat exchanger
529	85.0	42.7	10	After direct evaporative cooler
530	78.0	41.0	10	After heat exchanger
531	82.6	41.0	9	After direct evaporative cooler
532	76.7	39.6	9	After heat exchanger
533	82.2	39.6	8	After direct evaporative cooler
534	74.7	37.8	8	After heat exchanger
535	79.4	37.8	7	After direct evaporative cooler
536	72.9	36.2	7	After heat exchanger
537	77.0	36.2	6	After direct evaporative cooler
538	71.4	34.9	6	After heat exchanger
539	74.8	34.9	5	After direct evaporative cooler
540	69.9	33.7	5	After heat exchanger
541	72.9	33.7	4	After direct evaporative cooler
542	68.7	32.7	4	After heat exchanger
543	71.2	32.7	3	After direct evaporative cooler
544	67.5	31.8	3	After heat exchanger
545	69.7	31.8	2	After direct evaporative cooler
546	66.6	31.0	2	After heat exchanger
547	68.6	31.0	1	After direct evaporative cooler
548	66.0	30.4	1	After heat exchanger (exhaust)

TABLE 2

Primary Air Stream:

Location	Temperature (degrees F)	Enthalpy (Btu/lbm)	Stage	Location
600	74.0	30.2	1	Inlet
601	73.6	30.1	1	After heat exchanger
602	81.1	30.1	1	After desiccant
603	78.6	29.5	2	After heat exchanger
604	80.7	29.5	2	After desiccant
605	77.9	28.8	3	After heat exchanger
606	77.9	28.8	3	After desiccant
607	75.5	28.2	4	After heat exchanger
608	78.4	28.2	4	After desiccant
609	75.4	27.5	5	After heat exchanger
610	75.4	27.5	5	After desiccant
611	72.6	26.9	6	After heat exchanger
612	72.6	26.9	6	After desiccant
613	70.2	26.3	7	After heat exchanger
614	70.2	26.3	7	After desiccant
615	68.0	25.7	8	After heat exchanger
616	68.0	25.7	8	After desiccant
617	65.9	25.2	9	After heat exchanger
618	65.9	25.2	9	After desiccant
619	64.1	24.8	10	After heat exchanger
620	64.1	24.8	10	After desiccant
621	62.4	24.3	11	After heat exchanger
622	62.4	24.3	11	After desiccant
623	60.9	24.0	12	After heat exchanger
624	60.9	24.0	12	After desiccant (supply air)

TABLE 2-continued

Secondary Air Stream:			
Location	Temperature (degrees F)	Enthalpy (Btu/lbm)	Stage Location
625	73.2	36.0	12 After direct evaporative cooler
626	72.5	35.8	12 After heat exchanger
627	76.1	35.8	11 After direct evaporative cooler
628	71.2	34.7	11 After heat exchanger
629	75.1	34.7	10 After direct evaporative cooler
630	69.5	33.3	10 After heat exchanger
631	73.0	33.3	9 After direct evaporative cooler
632	68.2	32.1	9 After heat exchanger
633	72.3	32.1	8 After direct evaporative cooler
634	66.3	30.7	8 After heat exchanger
635	69.9	30.7	7 After direct evaporative cooler
636	64.5	29.4	7 After heat exchanger
637	67.8	29.4	6 After direct evaporative cooler
638	62.9	28.2	6 After heat exchanger
639	65.8	28.2	5 After direct evaporative cooler
640	61.4	27.2	5 After heat exchanger
641	63.9	27.2	4 After direct evaporative cooler
642	60.0	26.2	4 After heat exchanger
643	62.3	26.2	3 After direct evaporative cooler
644	58.7	25.4	3 After heat exchanger
645	60.7	25.4	2 After direct evaporative cooler
646	57.5	24.6	2 After heat exchanger
647	59.5	24.6	1 After direct evaporative cooler
648	56.7	23.9	1 After heat exchanger (exhaust)

FIG. 12 shows air temperatures for a system for cooling return air. This system is similar to that of FIG. 11 except for the different entering-air conditions. Table 2 describes each location. This system is capable of providing a supply air temperature of 61° F., which is sufficiently low to be compatible with most conventional air-distribution systems. The supply air is at about 75% relative humidity, which is sufficiently low to prevent mold growth in ducts and maintain comfortable space humidity.

These supply-air conditions are illustrative of what is possible with this system. Changes in the details of the design can change the supply air conditions to whatever is required. For example this system is suitable for use in an application which uses a 65 to 70° F. supply air temperature such as is described in my co-pending application entitled "High Efficiency Air Conditioning System with High Volume Air Distribution." Low temperature air is also possible, but with reduced performance.

A complete cooling system is a combination of the system in FIGS. 10 and 11. The system in FIG. 12 exhausts approximately half of the return air. The system in FIG. 11 supplies the make-up air required to replace this exhaust air.

#### High System Efficiency

The efficiency of these systems is quite high. For these systems the coefficient of performance (COP) is defined as cooling output divided by latent heat absorbed by the desiccant. For the system in FIG. 11, the COP is approximately 2.1. For the combined system including ventilation load the efficiency is about 1.5.

Note that this configuration requires the introduction of outside air. If the basis of comparison is system with no outside air, then there should be no credit for the load associated with cooling the outside air to the building conditions. On this basis the system COP is approximately 0.8.

If the desiccant is recharged using a solar collector, the collector efficiency must be considered. For the three-stage collector shown in FIG. 9, the waste heat from one stage is

used to drive the next, which theoretically can triple the output of concentrated desiccant solution. In real life, the collector loss would reduce the effect. Assuming a 50% collector loss, the total system efficiency based on solar input to cooling output can exceed 2.0. This performance is much better than what is possible with expensive absorption chillers with high-performance solar collectors, which gives a system efficiency of 0.2 to 0.5 at best. This efficiency advantage translates into a massive reduction in collector cost and area required to drive the new system compared to the prior art. Even with the use of low-cost collectors such shown in FIG. 7, the new system has a massive efficiency advantage compared with the prior art.

This high performance allows the use electric resistance as a back-up heat source. The efficiency with electric backup should be higher than that for a solar input, since transmission and reflection losses are not a factor. This means that system efficiency in the range of 2 to 3 based on electric input is possible. This efficiency is comparable to that of conventional electric air conditioners. If combined with a suitable storage system, electric back up can take advantage of inexpensive, off-peak electric rates. These rates can be as much as a factor of 10 lower than peak rates. The combination of solar input and off-peak rates can result in a massive reduction in energy cost compared to conventional systems.

#### Gas Turbine Inlet Cooler

FIG. 13 shows a system for cooling inlet air to a gas turbine that uses heat from the turbine for regenerating the desiccant solution. Ambient air 701 enters a regenerative desiccant cooler 700, which cools the air. The cooler includes pumps for circulating water and desiccant solution inside the cooler. The cooler receives make up water 710 and used water 711 drains from the cooler. Exhaust fan 714 draws the exhaust air 709 from the cooler and discharges away from the turbine to prevent recirculation. The turbine inlet air 702 leaves the cooler 700 and enters the gas turbine 713. Fan 715 adds ambient air 704 to the turbine exhaust air 703 to form mixed air 705. The mixed air 705 enters regenerator 706 where it evaporates water from the desiccant solution. The regenerator comprises an extended surface that is wetted with desiccant liquid. It may be made of materials similar to that used in direct evaporative coolers. Mixing ambient air with turbine exhaust lowers the temperature of air entering the regenerator, which allows the use of inexpensive low-temperature materials. Pump 716 moves diluted desiccant solution 708 from the cooler 700 to the regenerator 706. The regenerator can include a pump or circulating desiccant liquid inside the regenerator. Concentrated desiccant 707 returns to the cooler from the regenerator. Outlet air 712 exits from the regenerator.

This system can increase turbine output power by roughly 20 percent at summer peak conditions. The capacity gas turbine declines by about 0.4 percent per degree Fahrenheit. A 20% improvement in capacity corresponds to cooling the inlet temperature is reduced from 100 to 50° F. at peak conditions. The system can also control relative humidity to the turbine. Turbine efficiency improves by roughly 0.1%/°F. which corresponds to as much as 5% improvement at peak conditions. Input power for fans and pumps needed to operate the desiccant system is small and should not significantly effect these figures.

#### Heat-Exchanger Details

FIGS. 14a, 14b, and 14c show an alternate gas-to-gas heat exchanger design using paper and cardboard. Sheets of

paper **800** are supported between first spacers **801** and second spacers **802**. The spacers are preferably made of corrugated cardboard or similar material. As shown in FIG. **14b**, first spacers **801** are oriented to allow primary air stream **803** to flow the length of the paper in a single pass. 5  
Second spacers **802** are set to form multiple passes of secondary air stream **804**. The whole heat exchanger is coated with a material such as linseed oil, acrylic, wax, etc. which serves as both an adhesive and a protective coating. 10  
While this drawing shows a two-pass arrangement on the secondary side, similar geometries can accommodate any number of passes. This heat exchanger construction has many applications including exhaust-air heat recovery in addition to use in evaporative and desiccant systems.

#### Dehumidifier Embodiment

FIG. **15** shows another preferred embodiment that is acts as a dehumidifier. Desiccant fluid **900** is contained in an insulated container **904** and is heated by solar radiation that is transmitted through a cover **902**. Moisture evaporates from the fluid and forms condensate **906** on the bottom side of the cover. The condensate flows down the underside of the cover and collects in a channel **908**. The desiccant fluid moves by natural convection through channels **910** and **912** to a tank **918**. An air pump **914** blows air through a tube **915** into the desiccant fluid **900** forming bubbles **916**. The flow of air mixes the desiccant liquid in the tank. A fan **922** draws air over the desiccant fluid, which dehumidifies the air stream. A baffle **920** directs the air to toward the surface of the liquid. The baffle also acts to cut off air flow if the liquid level gets too high, thus preventing overflow of desiccant liquid. A level switch may also be included to turn off the air pump and fan at high liquid levels.

This embodiment may be useful as a small dehumidification system that can fit in a window. It may be especially useful for bathrooms or basements in homes.

Overall this invention has many advantages over the prior art:

- 1) Simple, low-cost, reliable designs
- 2) Counter cross flow configuration reduces cost and achieves good performance
- 3) Evaporative system with the ability to approach the dewpoint temperature
- 4) Desiccant system can act as a heat pump to raise supply air temperatures
- 5) Ability to use low-cost, safe, desiccants such as calcium chloride for air-conditioning
- 6) Compact energy, inexpensive, low-cost, energy-storage capability in the form of a concentrated desiccant solution
- 7) Low-cost solar regeneration
- 8) Ability to use off-peak electricity for back-up to solar
- 9) Ability to separate the wet portions of the system so as to allow easy replacement
- 10) Use of reliable, low-cost air-lift pumps which allow the use of many different liquid circuits
- 11) Extremely high efficiency possible
- 12) Simple controls
- 13) Ability to use readily available components
- 14) Low-cost paper-based heat exchangers

What is claimed is:

1. A solar air conditioner comprising,

- a. a solar collector that comprises a collector surface that is oriented to receive thermal energy from the sun and a means for spreading desiccant fluid over said collector surface so that said solar radiation warms said desiccant fluid and evaporates water from it,

- b. a mass-transfer device that transfers moisture from a primary air stream to said desiccant fluid,
- c. cooling means that is in contact with said primary air stream and that is capable of cooling said primary air stream to a temperature that is lower than the ambient wet-bulb temperature, and
- d. a fluid circuit that includes said mass transfer device and said solar collector so that moisture moves from said primary air stream into said desiccant fluid in said mass-transfer device; desiccant fluid moves from said mass transfer device to said solar collector which uses solar energy to evaporate water from the desiccant fluid; and concentrated desiccant fluid returns to said mass-transfer device; and said cooling means lowers the temperature of said primary air stream so as to cool and dehumidify said primary air stream.

2. The solar air conditioner of claim **1** further comprising a secondary air stream and wherein said cooling means comprises:

- a. a heat-exchange device that transfers thermal energy from said primary air stream to said secondary air stream and
- b. means for evaporating water into said secondary air stream before it exits said heat-exchange device so as to cool the secondary air stream and thereby reduce the temperature of said primary air stream.

3. The solar air conditioner of claim **2** wherein said secondary air stream is drawn from air that has been previously conditioned by said solar air conditioner.

4. The solar air conditioner of claim **3** wherein said primary air stream is drawn from the ambient air.

5. The solar air conditioner of claim **4** wherein said secondary air stream is exhausted to the ambient after passing through said heat exchanger.

6. The solar air conditioner of claim **5** wherein said heat-exchange device comprises at least two stages that are arranged in an approximately counterflow configuration so that a first stage removes thermal energy from primary air before it enters a second stage and adds thermal energy to said secondary air after it exits said second stage.

7. The solar air conditioner of claim **6** wherein said mass-transfer device comprises at least two stages that are configured so that at least one stage of said heat-exchange device removes thermal energy from said primary air stream after it enters said first a first stage of said mass-transfer device and before said air stream enters a second stage of said mass-transfer device.

8. The solar air conditioner of claim **7** wherein said at least one stage of said heat-exchange device transfers thermal energy from said primary air stream before it enters said first stage of said mass-transfer device.

9. The solar air conditioner of claim **1** further comprising an auxiliary source of thermal energy that heats said desiccant liquid so that water vapor may evaporate from said desiccant liquid when solar energy is insufficient.

10. The solar air conditioner of claim **9** wherein said auxiliary source of thermal energy comprises an electric heater that is connected by a thermally conductive pathway to said collector surface so as to heat desiccant fluid flowing over said collector surface.

11. The solar air conditioner of claim **9** further comprising a channel carrying a hot fluid that is connected by a thermally conductive pathway to said collector surface as to heat desiccant fluid flowing over said collector surface.

12. The solar air conditioner of claim **1** wherein said means for spreading desiccant fluid over said collector surface comprises means for trickling said desiccant fluid over said collector surface.

## US 6,513,339 B1

17

13. The solar air conditioner of claim 1 wherein said means for spreading desiccant fluid over said collector surface comprises means for pooling said desiccant fluid over said collector surface.

14. The solar air conditioner of claim 1 wherein said solar collector further comprises a first light-transmissive cover that can transmit solar radiation located above said collector surface so as to prevent precipitation from the atmosphere from diluting said desiccant fluid.

15. The solar air conditioner of claim 14 wherein said solar collector further comprises a first enclosure that is bounded at the top by said first light-transmissive cover and at the bottom by said collector surface so as to prevent movement of gas between said enclosure and the ambient and wherein said first light-transmissive cover is oriented at an angle with respect to horizontal so as to allow water that condenses on the bottom surface of the light-transmissive cover to drain without dripping onto the collector surface.

16. The solar air conditioner of claim 15 further comprising a second light-transmissive cover located over said first light-transmissive cover and comprising means for trickling desiccant fluid over the surface of said first light-transmissive cover so that heat transmitted through said first light-transmissive cover evaporates water from said desiccant fluid flowing over the first light-transmissive cover.

17. The solar air conditioner of claim 16 wherein said second light-transmissive cover is oriented at an angle with horizontal to allow water that condenses on the bottom surface of the second light-transmissive cover to drain without dripping into the desiccant fluid and further comprising:

- a. a third light-transmissive cover located over said second light-transmissive cover,
- b. means for trickling desiccant fluid over said the surface of said first second light-transmissive cover, and
- c. a second enclosure that is bounded at the top by said second light-transmissive cover and at the bottom by said first light-transmissive cover so that heat transmitted through the second light-transmissive cover evaporates water the desiccant fluid flowing over the second light-transmissive cover.

18. The solar air conditioner of claim 1 further comprising means for storing concentrated desiccant fluid sufficient for several hours of operation of the air conditioner.

19. The solar air conditioner of claim 1 wherein desiccant fluid moves through said fluid circuit by natural convection.

20. The solar air conditioner of claim 1 wherein said fluid circuit further comprises a pump for circulating desiccant fluid.

21. The solar air conditioner of claim 20 wherein said pump is an air-lift pump.

22. A heat-exchange device comprising:

- a. a first heat exchanger for transferring thermal energy between a two gas streams,
- b. a primary gas stream which flows through a first side of said heat exchanger,
- c. a secondary gas stream which flows through multiple passes on a second side of said heat exchanger in an

18

approximately counterflow configuration with respect to said primary gas stream and which comprises a first portion of said primary gas stream that exits said heat exchanger,

- d. a supply gas stream that comprises a second portion of said primary gas stream that exits said heat exchanger,
- e. a first fluid, and

f. a mass transfer means between said first fluid and said secondary gas stream whereby evaporation of said first fluid into said secondary gas stream lowers the temperature of said secondary gas stream, which allows thermal energy to move from said primary air stream to said secondary air stream through said heat exchanger and thereby reduce the temperature of the supply gas stream.

23. The heat exchange device of claim 22 wherein said first fluid is water.

24. The heat exchange device of claim 23 wherein said mass-transfer means comprises evaporative media that are in located between said passes of secondary air so that secondary air is alternately cooled by evaporating water from said evaporative media and warmed by the primary gas stream in a pass of said heat exchanger.

25. The heat exchange device of claim 24 further comprising a desiccant located in contact with the primary air stream.

26. The heat exchange device of claim 25 wherein said desiccant is a liquid desiccant and wets a mass-transfer surface.

27. An air conditioner that is capable of storing sensible and latent cooling comprising:

- a. a primary air stream,
- b. a secondary air stream that comprises air that was previously cooled by said air conditioner,
- c. a desiccant fluid,
- d. means for storing a quantity of desiccant fluid sufficient for several hours of operation,
- e. a regenerator that heats said desiccant fluid so as to evaporate water from said desiccant fluid,
- f. a mass-transfer device,
- g. a fluid circuit formed by said storage tank and said mass transfer device and said regenerator,
- h. a heat-exchange device that transfers thermal energy from said primary air stream to said secondary air stream,
- i. and means for evaporating water into said secondary air stream before it exits said heat-exchange device so as to cool the secondary air stream.

28. The device of claim 27 wherein said desiccant fluid comprises an aqueous solution of calcium chloride.

29. The device of claim 27 wherein heat for said regenerator is provided by off-peak electricity.

30. The device of claim 27 wherein heat for said regenerator is provided by solar energy.

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